AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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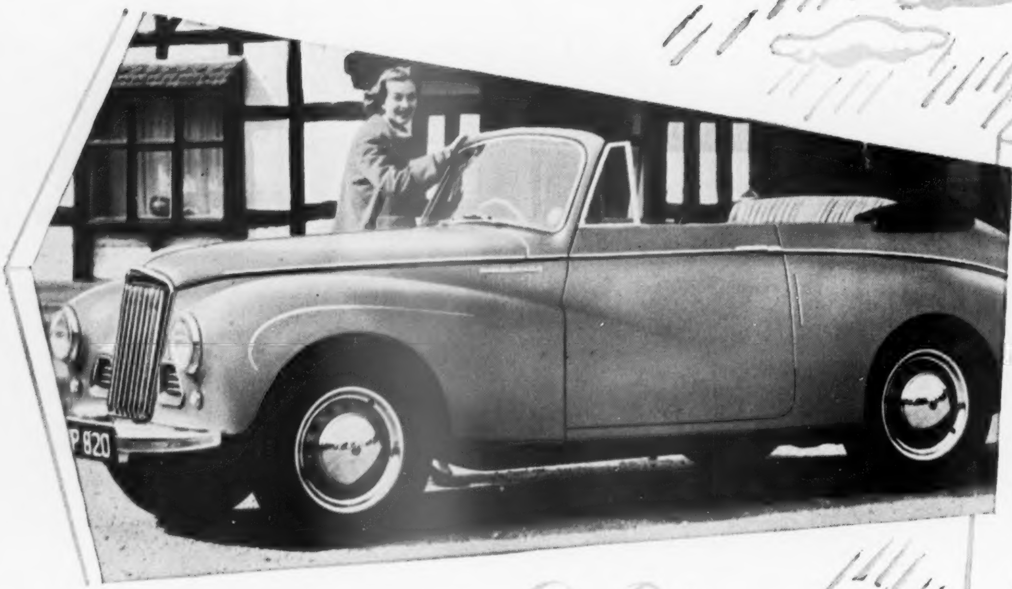

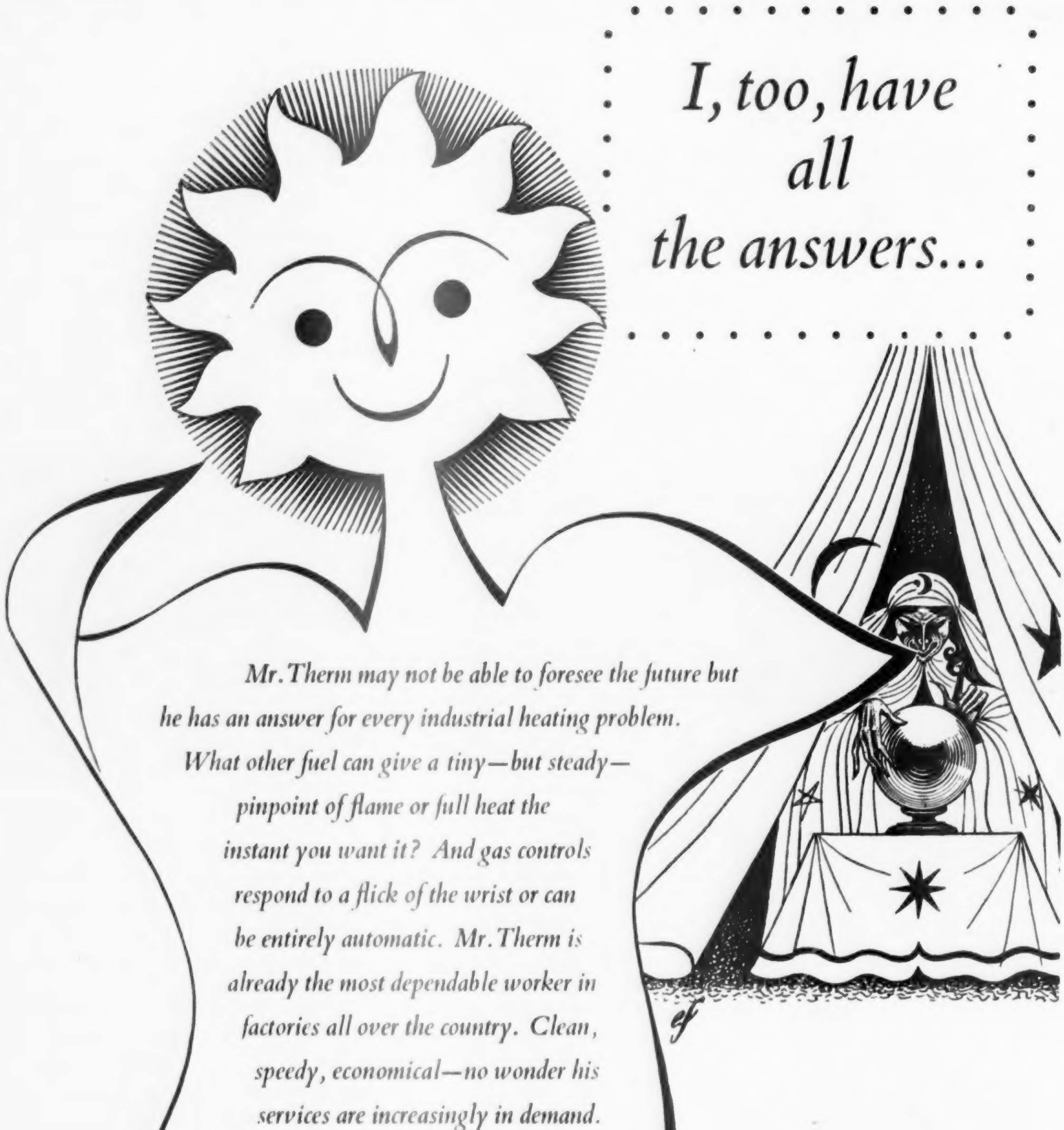

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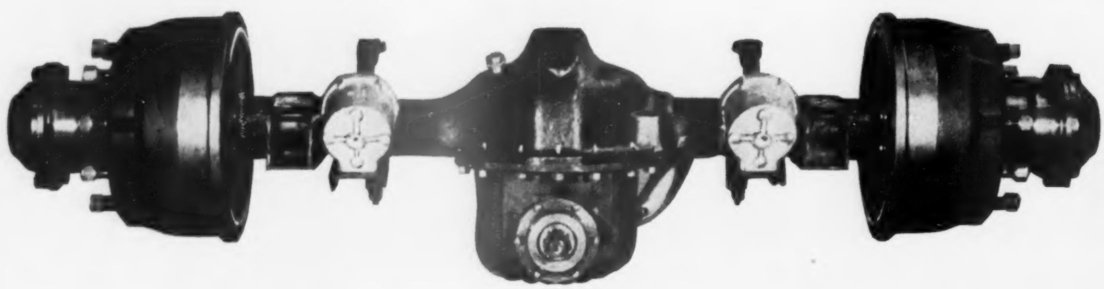
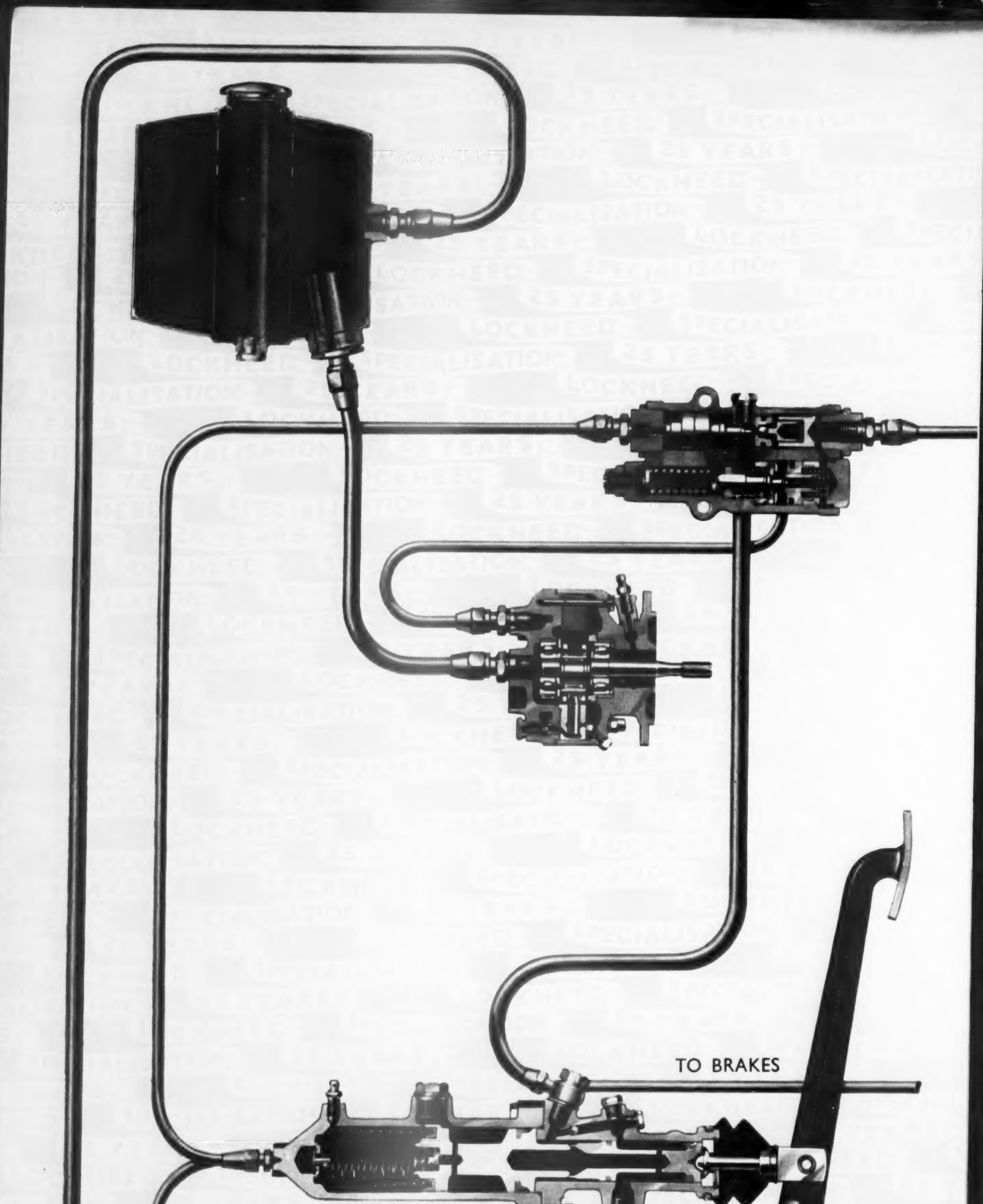
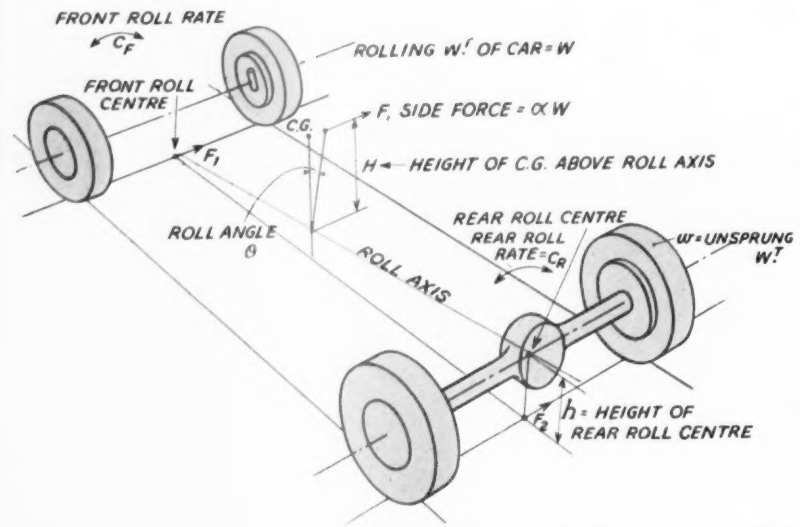



TO BRAKES
TO BRAKES

SUBTLETIES OF STEERING

Causes of over- and under-steer: 4. Car properties (a)



This diagram, similar to that in the February 1947 number of the Journal of the Institution of Automobile Engineers, explains the meaning of many of the terms used in the text.

DIFFERENT cars produce different steering behaviour from the same tyres because while cornering they impose different vertical and sideways loads upon the four tyres concerned. Suppose we consider first the way in which the weight carried by each wheel is affected by the car properties at a given sideways acceleration or severity of cornering.

Barring a minor correction which we shall mention and introduce later, the total amount of weight transference, at front and rear, from the inside to the outside wheels on a corner depends only on the sideways acceleration (or severity of the cornering), the height of the centre of gravity and the track. The method of finding wheel reactions by drawing the resultant through the c.g. and taking moments about each of the extremities of the supporting base in turn is well known: the supporting base is the quadrilateral contained by the four lines joining the four contact 'points' of the four tyres. This, however, tells us nothing as to how much of the extra load carried by the two outer tyres is taken by each. To find this out we must go rather more carefully into what happens when a car rolls outward on a corner, and how much it does so.

The conception of a roll centre at the front and the rear suspension is by now familiar to most of us, and the method of finding their positions by experiment (Olley, Proc. I.A.E. Vol. XXXII figs. 15, 16, and 17, Plates XII and XIII), by photographing the car, painted with a white line each end, at different roll angles from front and rear, using a still camera, applying pure rolling torque and taking multiple exposures on each plate or film, is also well known. The intersection of the various white lines showing on the photos giving the rolling pivot points at the fore and aft positions of the two white lines. A little further thought will show us that in fact the car is rolling about an axis passing through the two centres we have discovered in this way, and that the height of the intersections of this roll axis with the two vertical planes embracing the axes of the front and rear wheels gives the roll centre heights for front and rear suspensions.

Now each suspension has an anti-roll stiffness about its particular roll centre, and the sum of the two anti-roll stiffnesses, front and rear, is the anti-roll stiffness of the car. The centre of gravity of the sprung part of the car lies at a determinable distance above the roll axis. At a given sideways acceleration the sideways force due to that sideways acceleration has a moment about the roll axis, and this moment divided by the combined anti-roll stiffnesses of the two suspensions gives the roll angle of the car for those conditions. The anti-roll stiffness concerned is that about the ground, i.e. the tyre deflections must be taken into account. Allowance must also be made for the overhanging effect of the c.g. as the car rolls, which will slightly increase the rolling moment and hence the roll angle (this is the minor effect previously mentioned).

Knowing the roll angle and the anti-roll stiffnesses at the front and rear suspensions we know how much of the rolling couple is taken at each end of the car, and this knowledge combined with the track at front and rear gives us the weight transference from this cause at front and rear. This weight transference is only a part of the total: it is only that due to the moment of the c.g. about the roll axis and not about the ground. The remaining part of the weight transference is a separate matter which must be separately studied.

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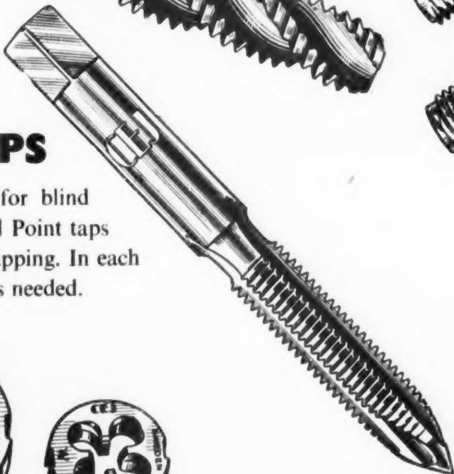

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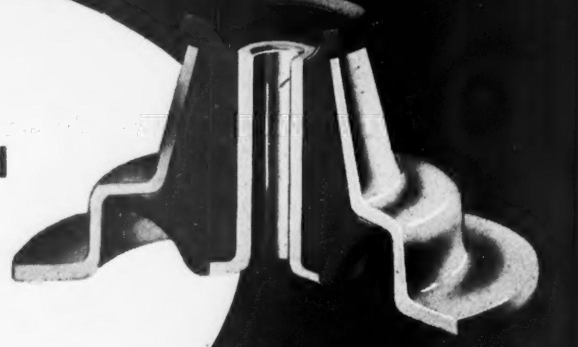
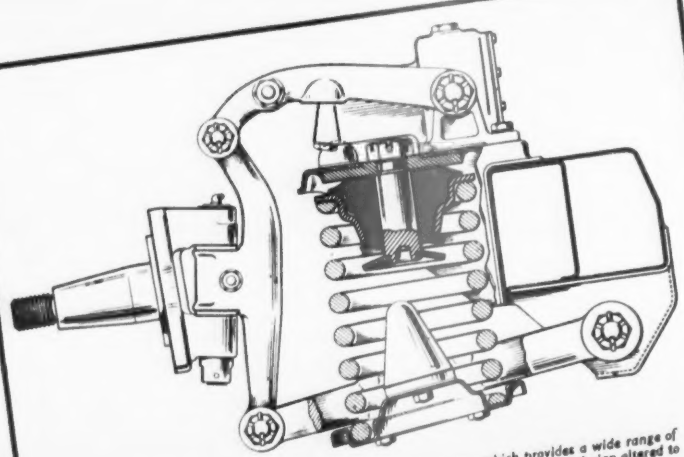


The 'Frustacon' mounting fitted to the I.F.S. of the 1952 Rover '75'. Sectioned to show rubber.



Silentbloc "Frustacon" fitted in an I.F.S. System. The mounting, which provides a wide range of deflection rates, may be fitted at the top or bottom of the coil spring and its design altered to suit any system.

To minimise the transmission of road noise to the body by the coil springs, Silentbloc Frustacon mountings have been incorporated in the I.F.S. of the 1952 Rover '75'. This notable step forward in the reduction of noise is another example of co-operation between car manufacturers and Silentbloc Ltd.

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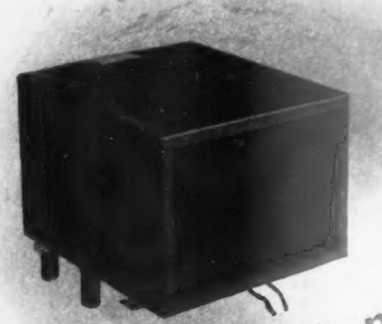
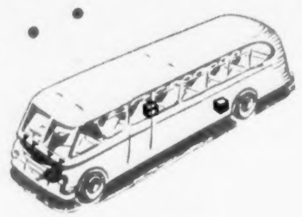
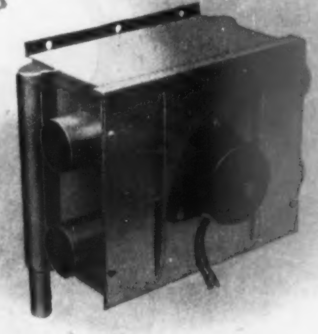


and screen demisting . . . . .





highly efficient

The 'S.12' heater with its high heat output is the ideal heating unit for any type of passenger vehicle. It can be used singly or for the larger coaches two or three units may be evenly disposed in the saloon, depending of course on the size and design of the vehicle and operating conditions required.

The 'S.8' demister unit has four separate high velocity outlets, and is the simplest method of preventing mist and ice formation on the driver's screens and windows.

light and compact

Completely self-contained and fully enclosed, both units are compact and of lightweight construction. Easy access is provided for cleaning the element and servicing the motor.

easily installed

The 'S.12' heater is secured by four screws to the vehicle floor, the water connections being under floor. It may be situated at the front of the gangway or in under-seat positions. No ducting is required.

The 'S.8' demister is normally mounted in the panelling beneath the windscreen and centrally disposed to give a simple piping arrangement to the demister nozzles. Water connections are brought up from the under-floor piping layout.

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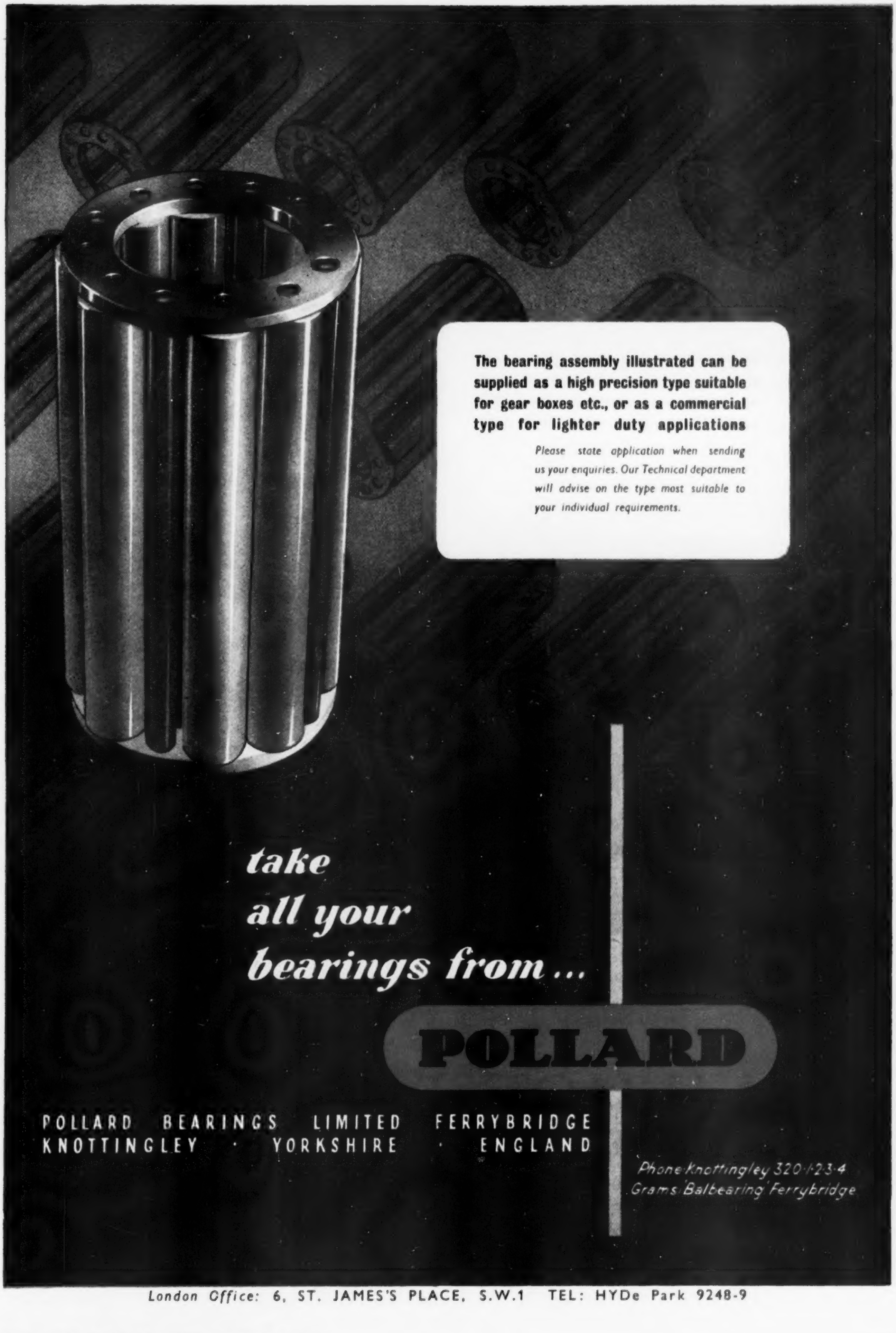

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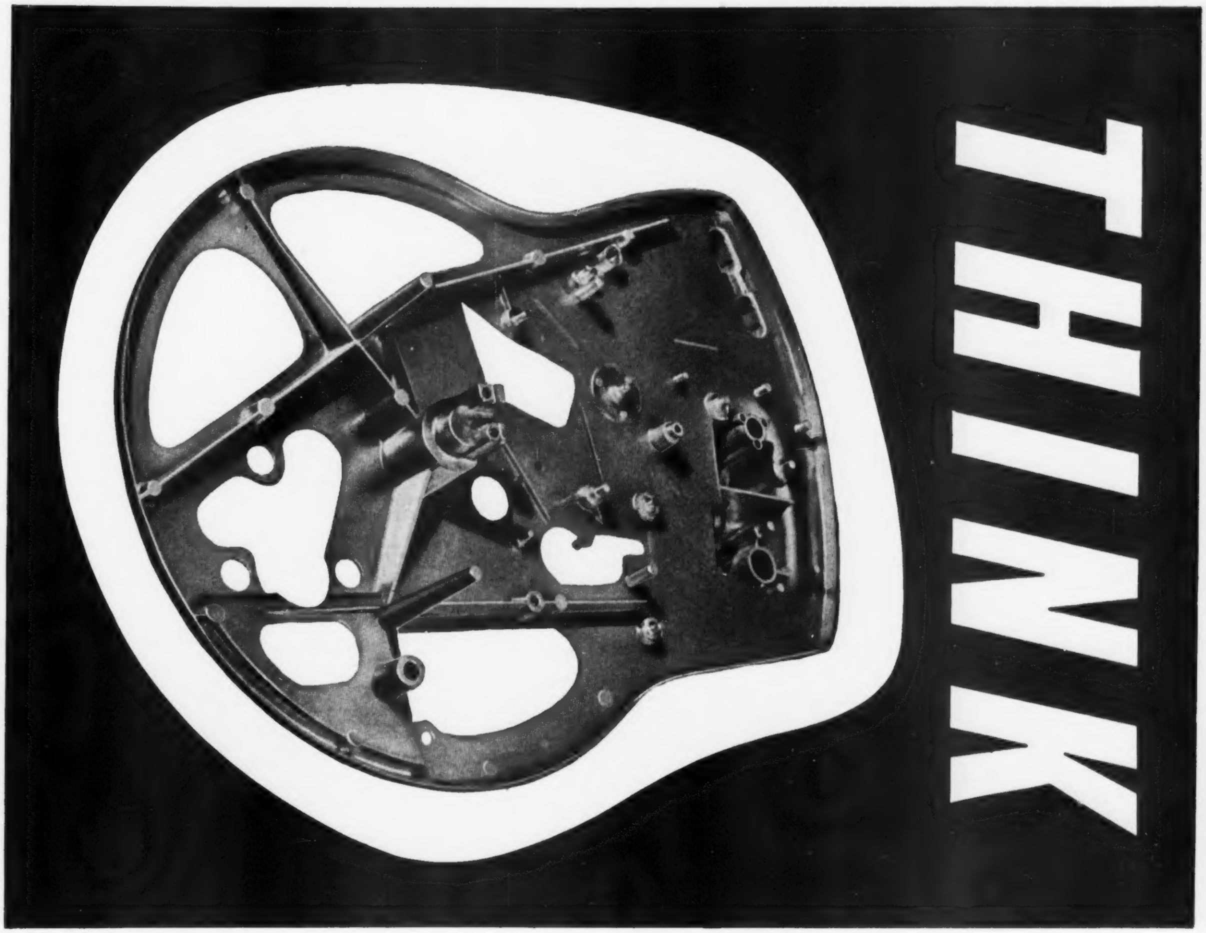

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Almost as quickly as you can say the words " Think Zinc," molten zinc alloy has become the Plessey gramophone record-changer base you see here. This speed in production is just one reason why in recent years so many industries have adopted the zinc alloy die casting process. Other outstanding advantages are :—

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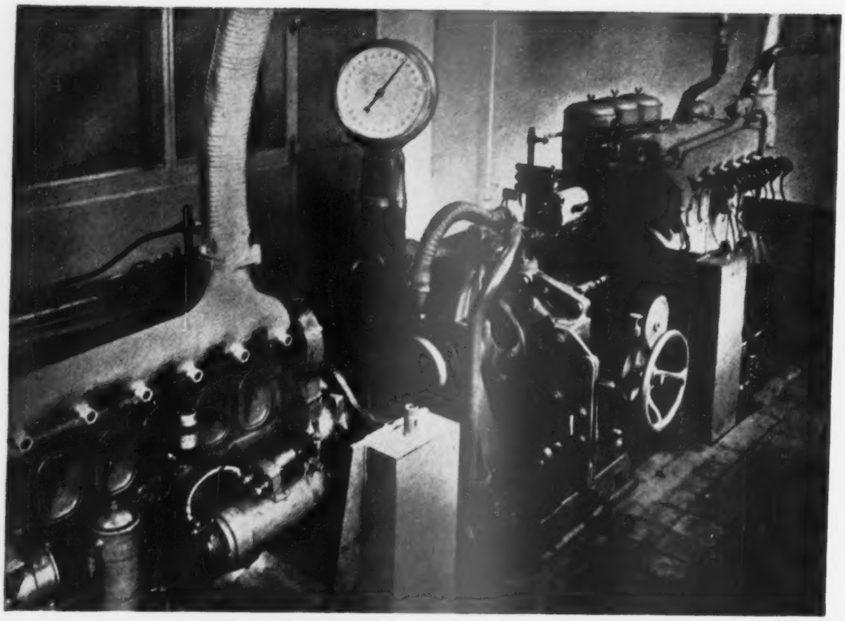

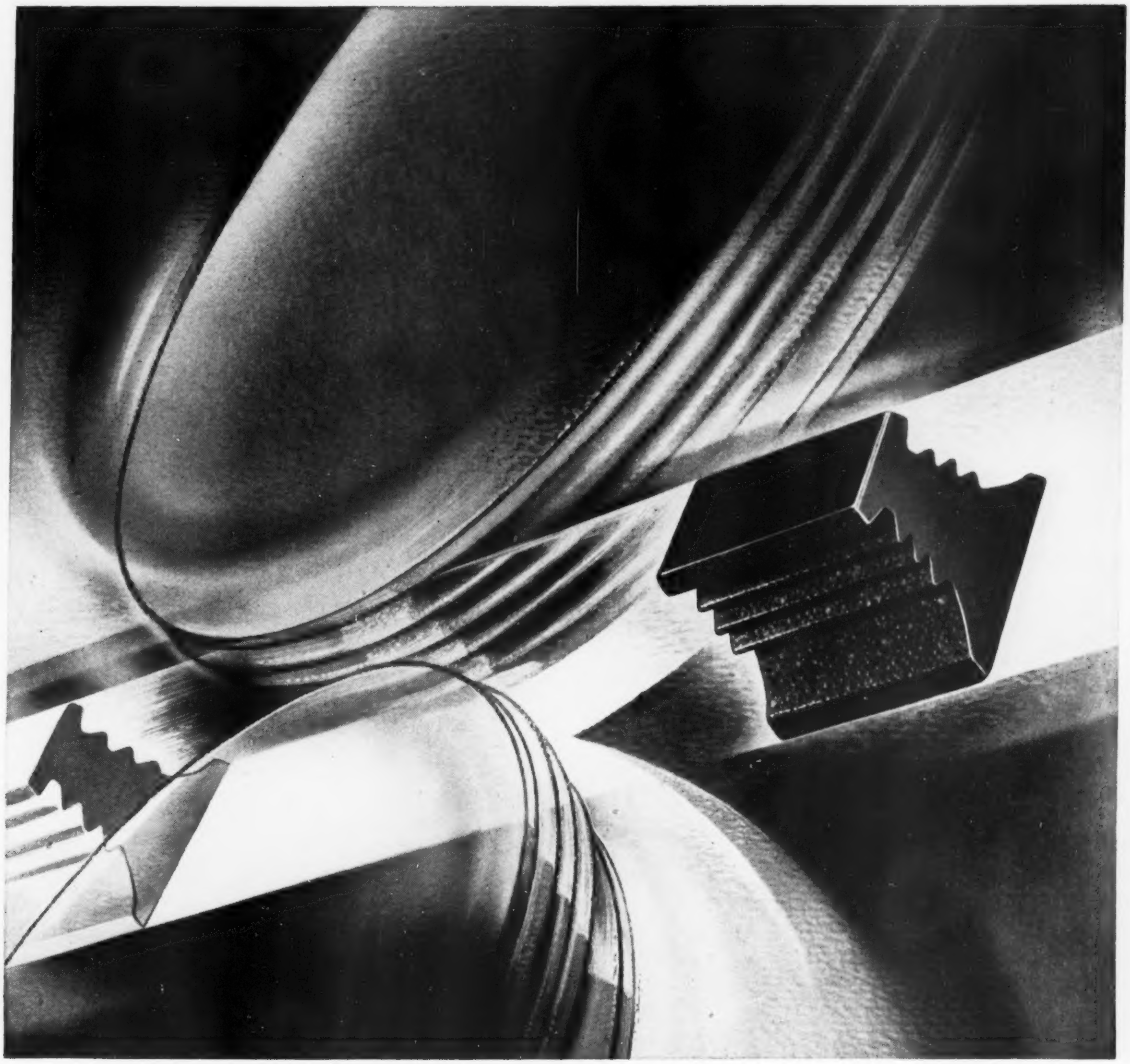

first again

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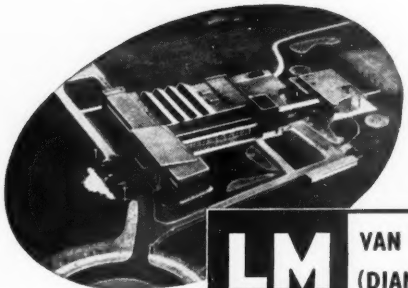

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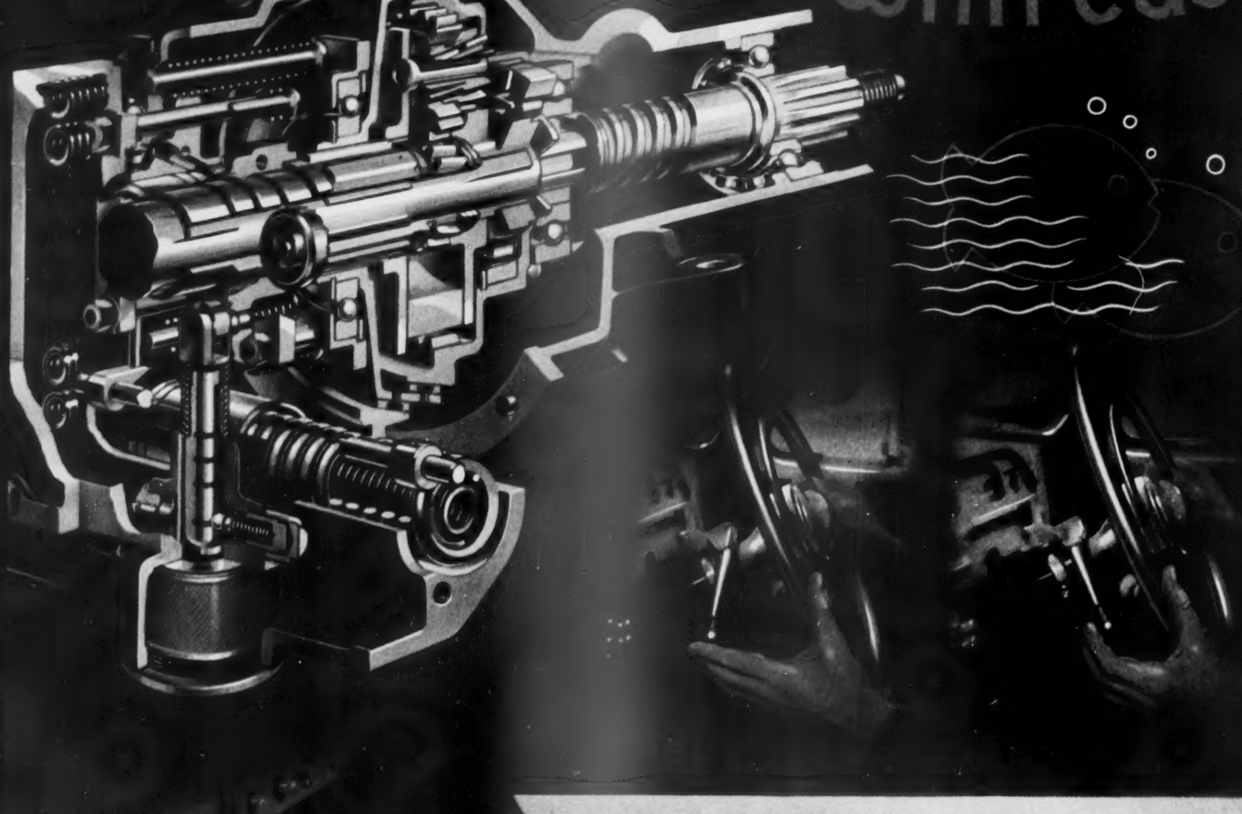


The LAYCOCK-de NORMANVILLE OVERDRIVE is a self-contained unit fitted to the rear of the gearbox (Fig. 1). It provides the so-much-desired extra high gear. (On a well-known make of car with a top gear ratio of 4.625 to 1, the overdrive raises it to 3.6 to 1.)

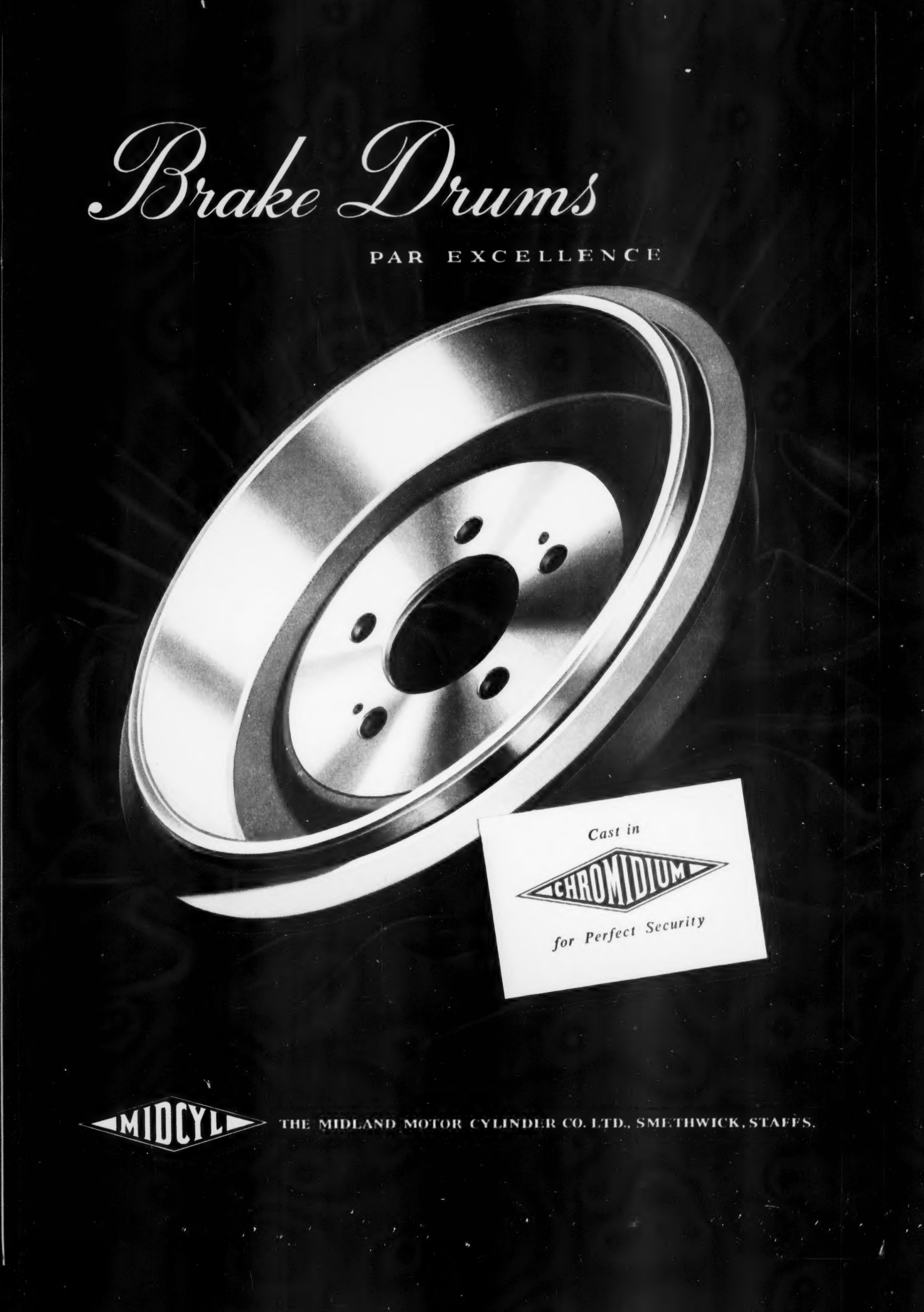
On this particular make of car the overdrive is engaged instantaneously by the mere finger-tip movement of the gear lever, from top to overdrive (Fig. 2 shows normal top gear position, Fig. 3 overdrive), and without movement of foot controls (Fig. 4). Full power is continuously transmitted, the operation being immediate, effortless and undetectable.

Control of this overdrive can be arranged in a variety of ways, and by co-operation with Joseph Lucas Ltd., a semi-automatic form of overdrive control has been developed. In order that the driver shall still have final control a manual switch is included in the circuit. Normally, with the manual switch not operating, at a pre-determined speed the car will change automatically into overdrive from top gear. The driver can at any speed (above cut-in speed) re-engage top gear by operating the manual switch, and this can be so arranged that his hand need not leave the steering wheel. By means of the Laycock-de Normanville Overdrive with the Lucas Automatic Overdrive Control an extra gear is added to the car, requiring no action on the part of the driver to engage it.

Technical leaflet describing the complete operation of this overdrive will be gladly mailed you on application to: Laycock Engineering Ltd., Victoria Works, Millhouses, Sheffield, 8.

The Laycock-de Normanville Overdrive

LAYCOCK ENGINEERING LTD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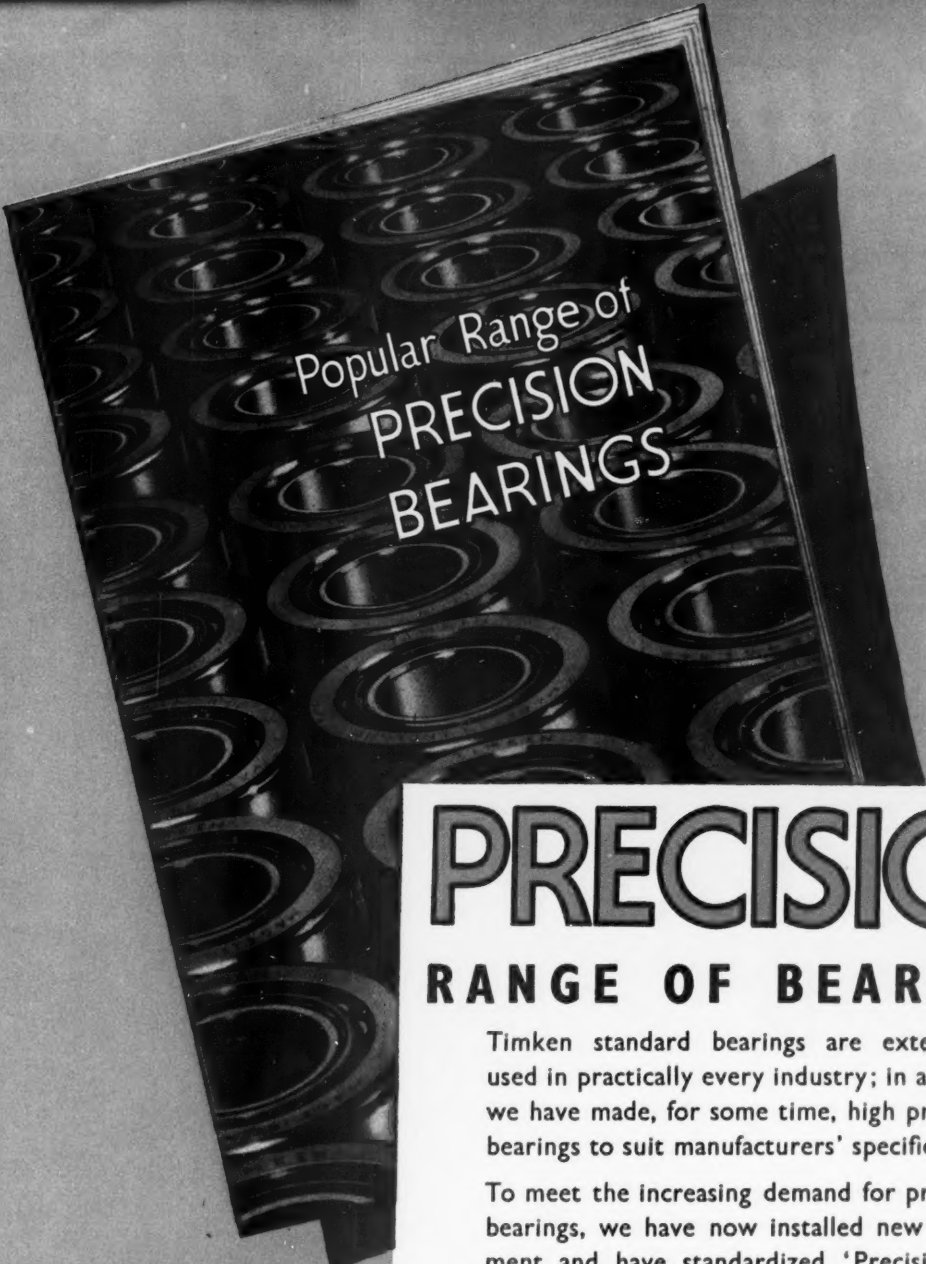

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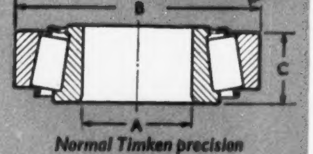
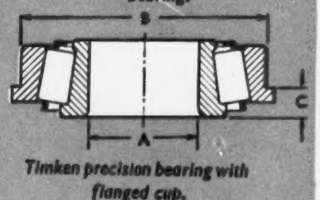


Normal Timken precision bearing.



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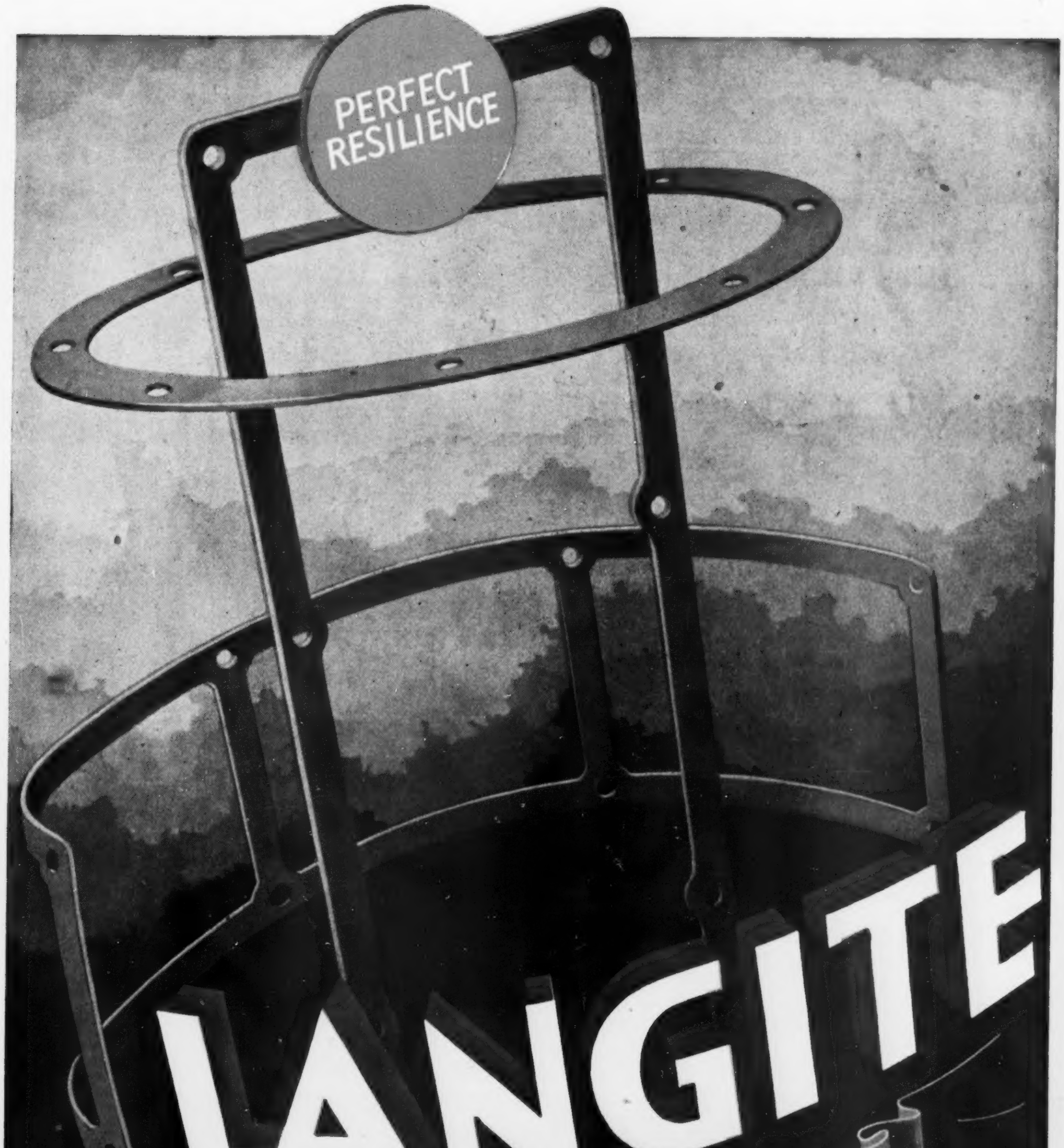
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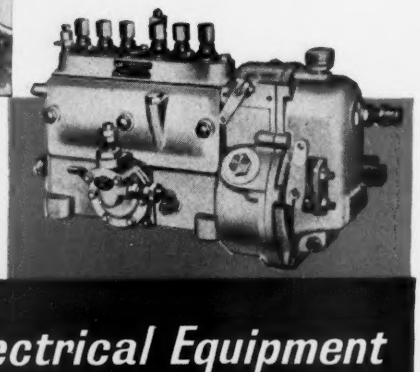

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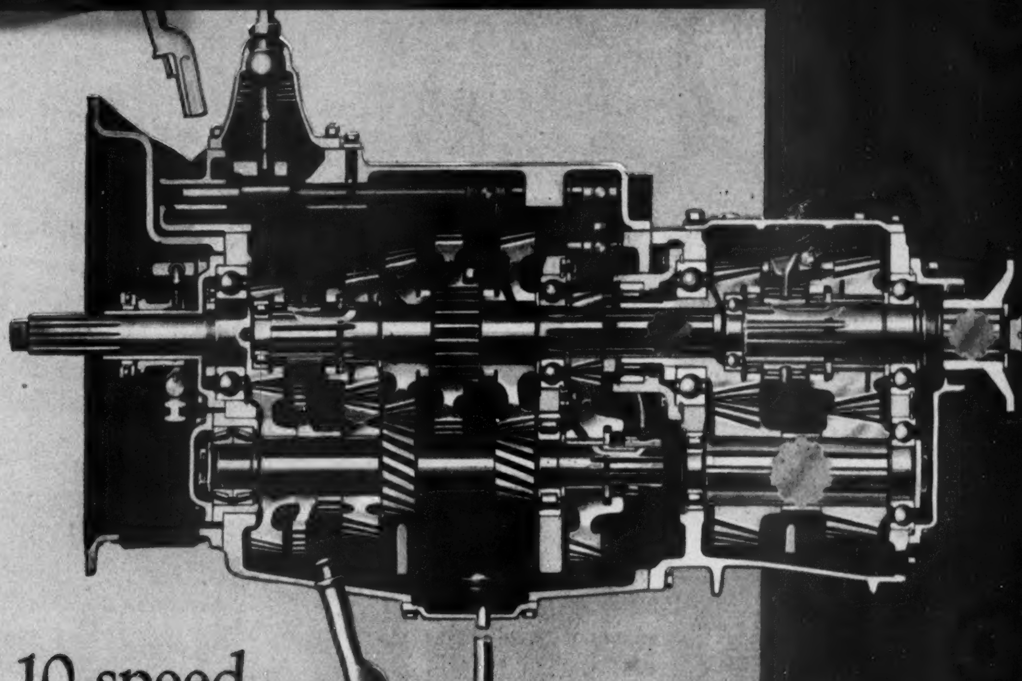
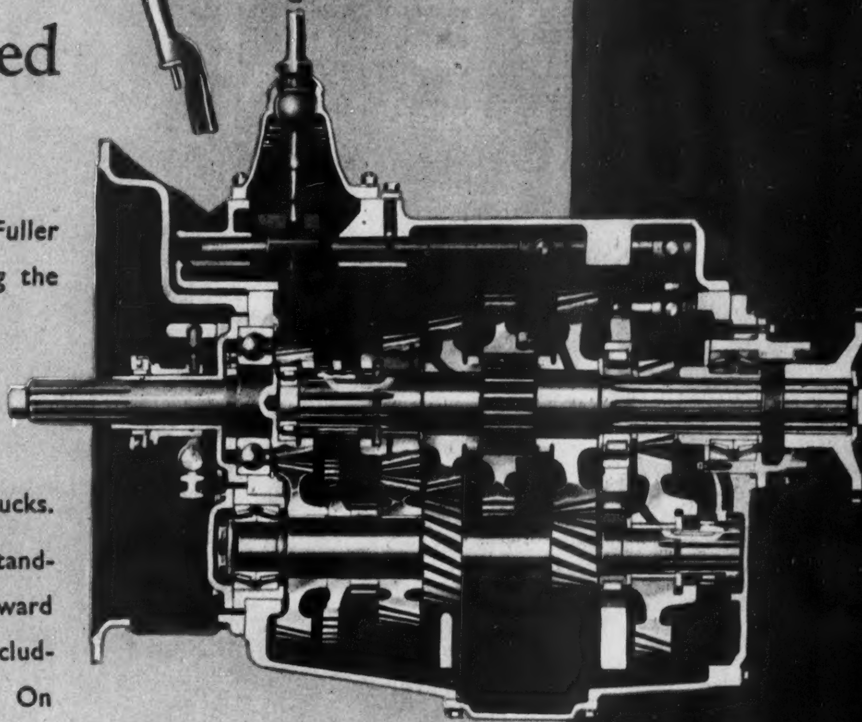


5 and 10-speed gear-boxes



The top illustration shows the Fuller 10-speed gear-box, comprising the famous five-speed box with a Fuller two-speed auxiliary box built on to it, thus providing a ten-speed box for heavy-duty operation on large trucks.

This unit is to the usual Fuller standard of high-duty, with all forward gears helical, and all changes, including reverse, by dog-clutches. On both of these boxes the gears are shot-peened and crown-shaved, to avoid stress concentration.

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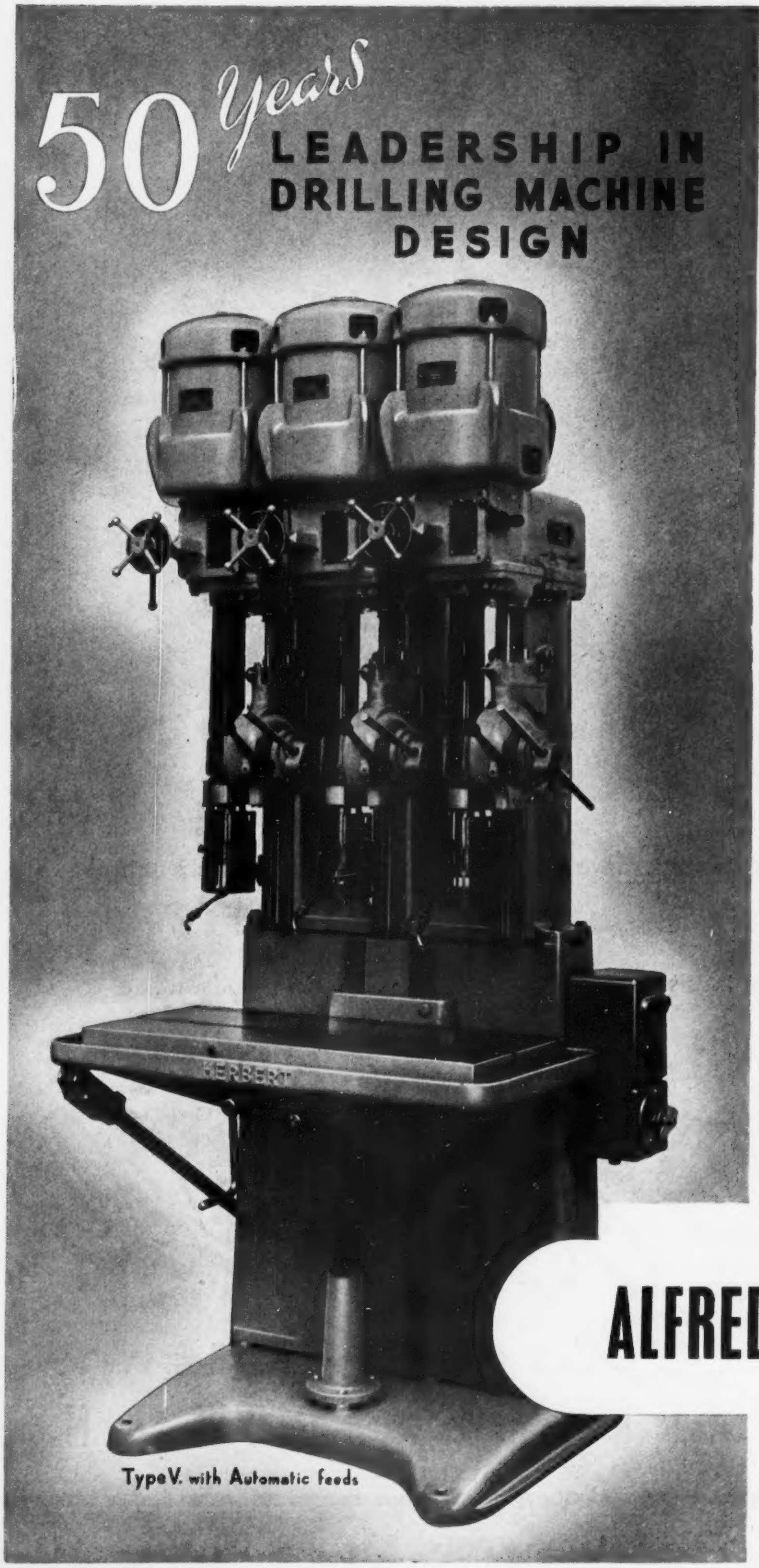

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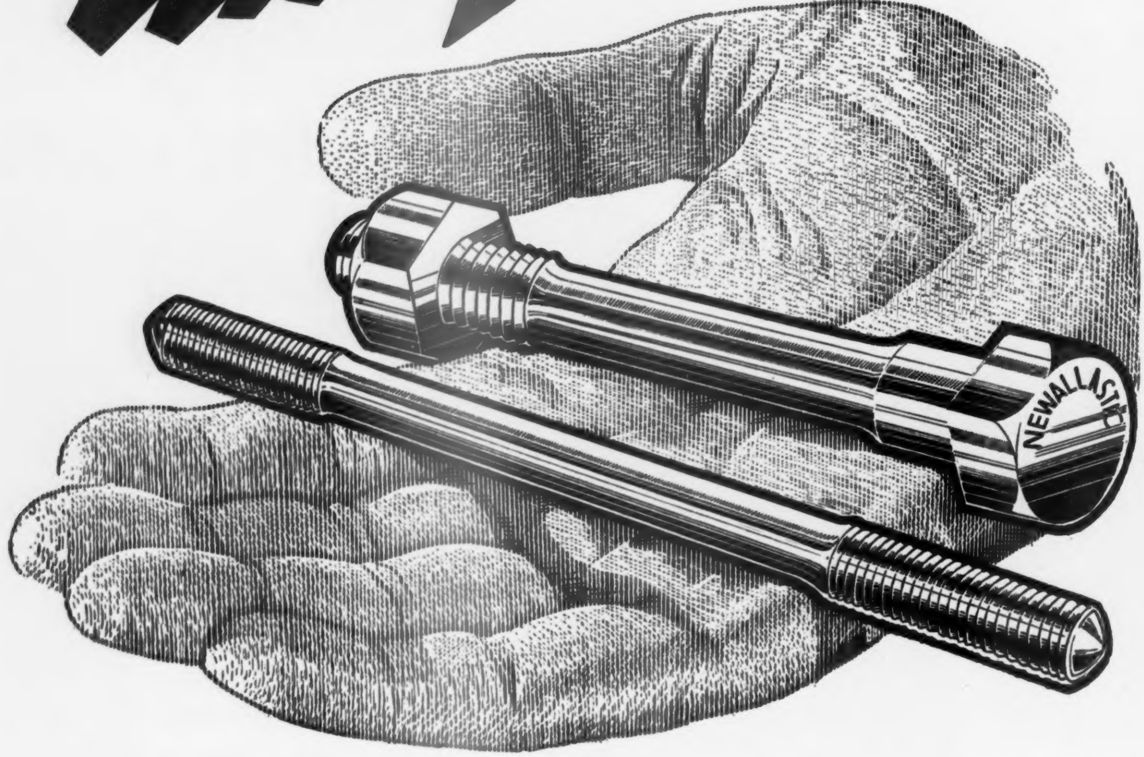

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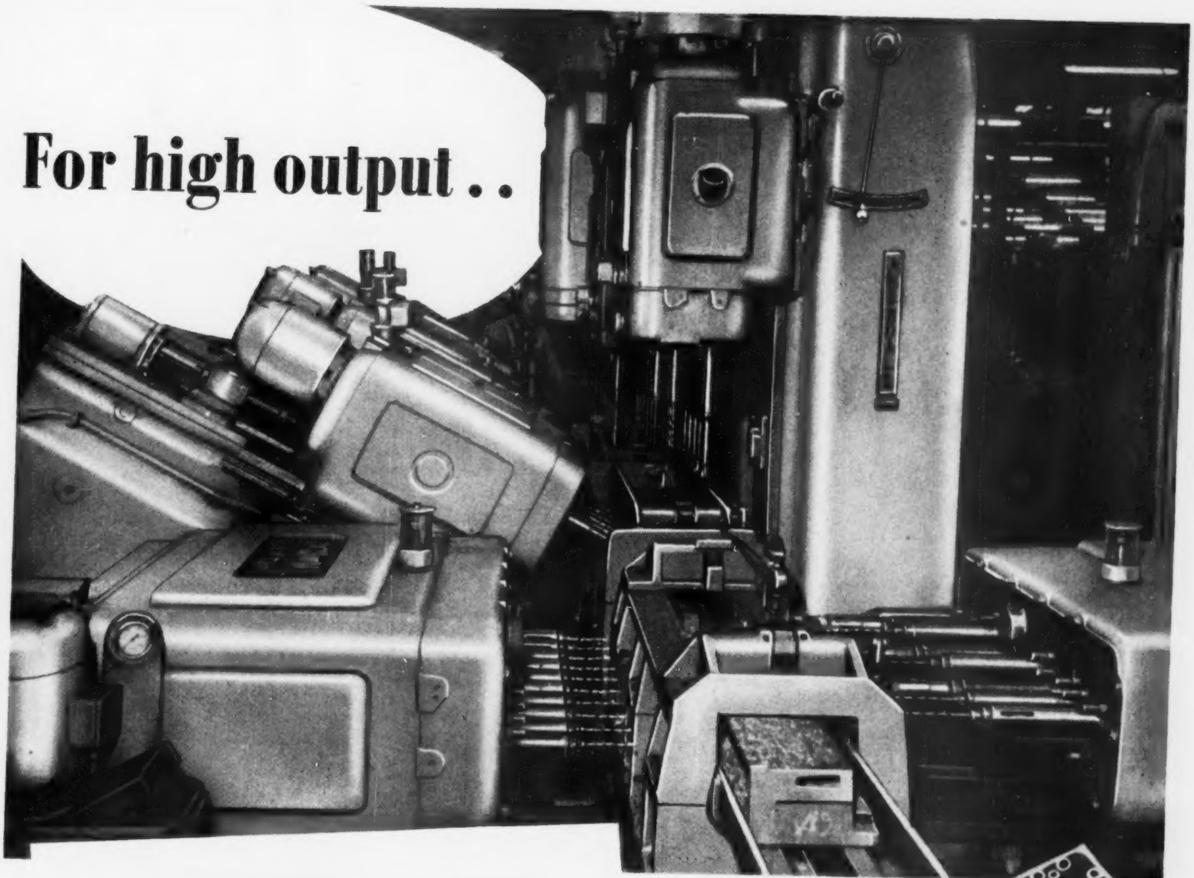

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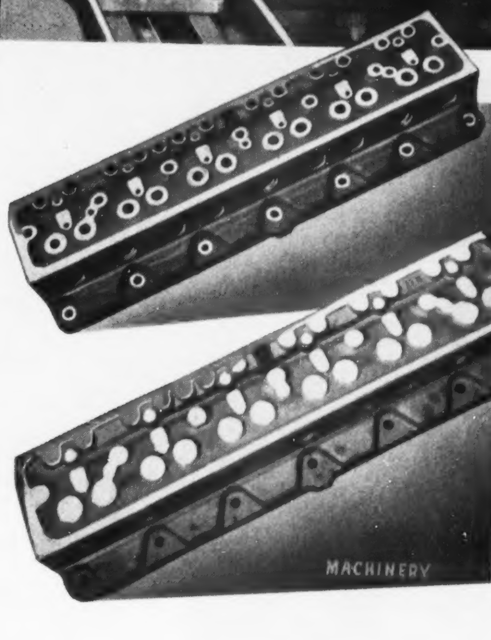

By courtesy of VAUXHALL MOTORS LTD., LUTON, we illustrate a special ARCHDALE machine for performing drilling, reaming, counter-boring and tapping operations on cylinder heads in an automatic, continuous sequence.

Developing such machines is a speciality of the ARCHDALE organisation, and the machine illustrated is only one of hundreds of special machines, including transfer machines, supplied to the automobile industry.

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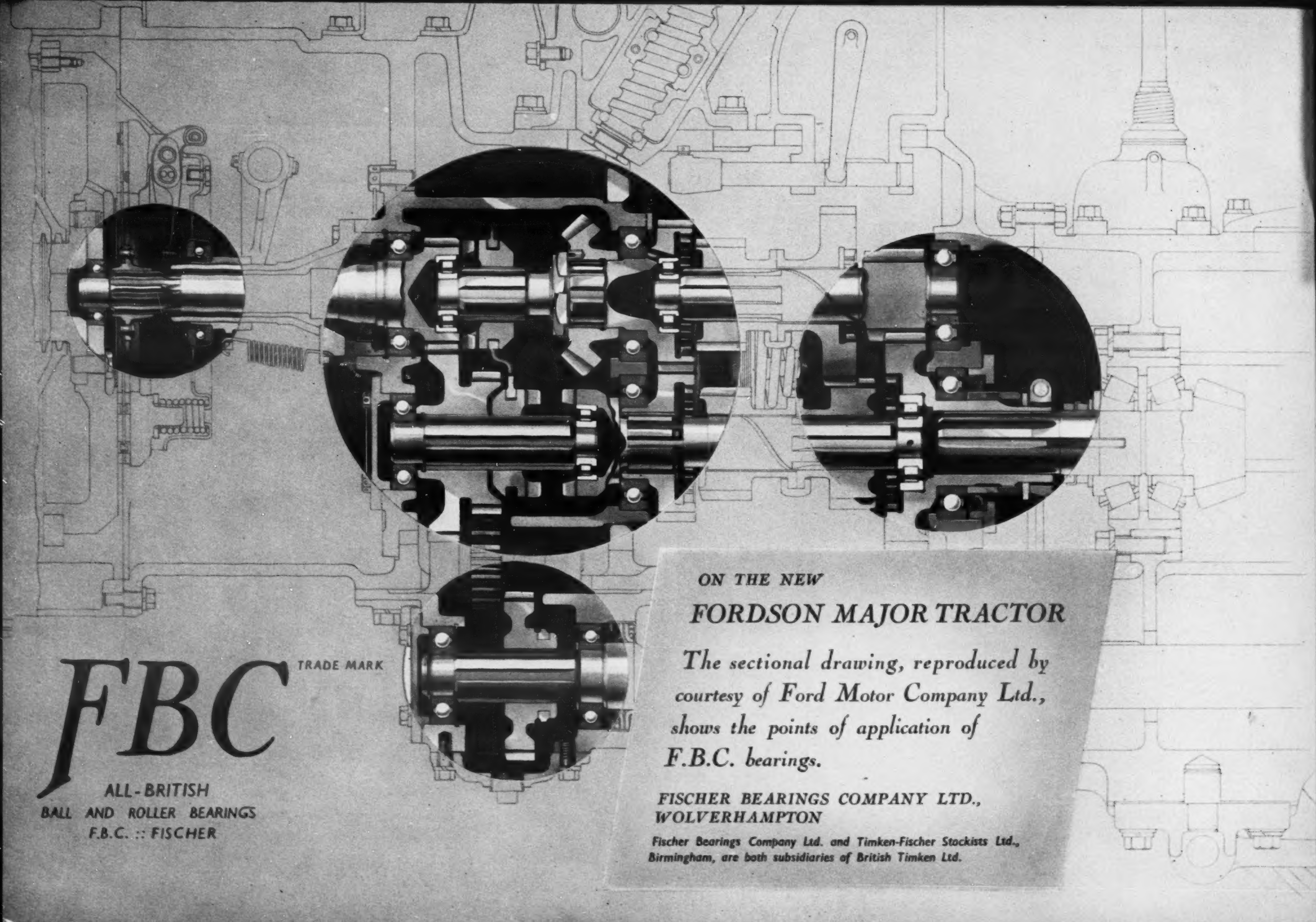
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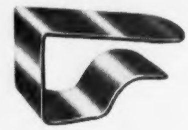
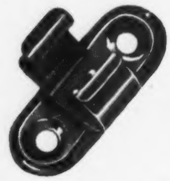


257 — a useful clip in black enamel, from ¾" to 1½".



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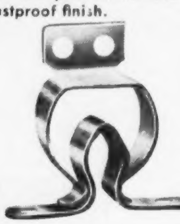

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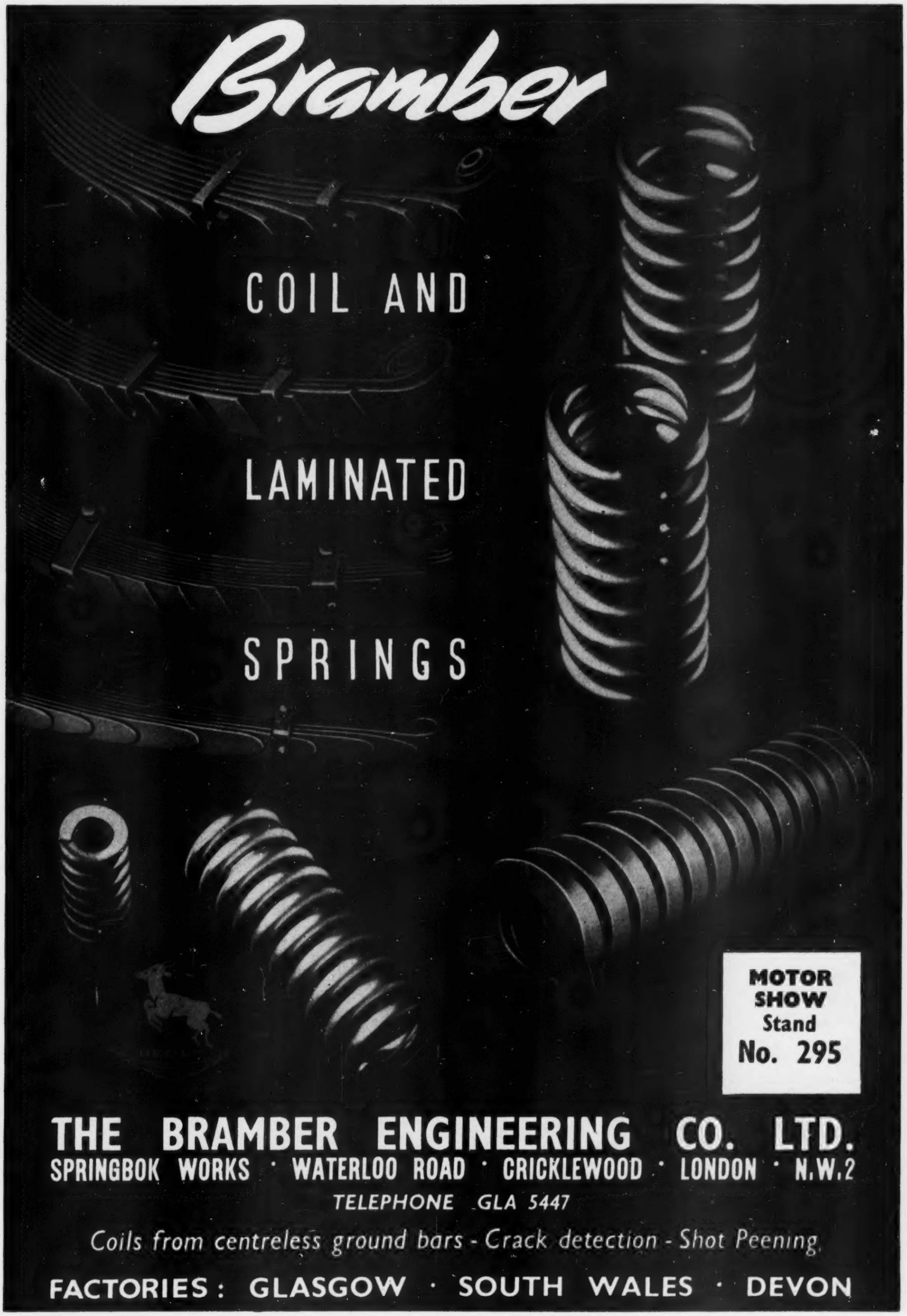
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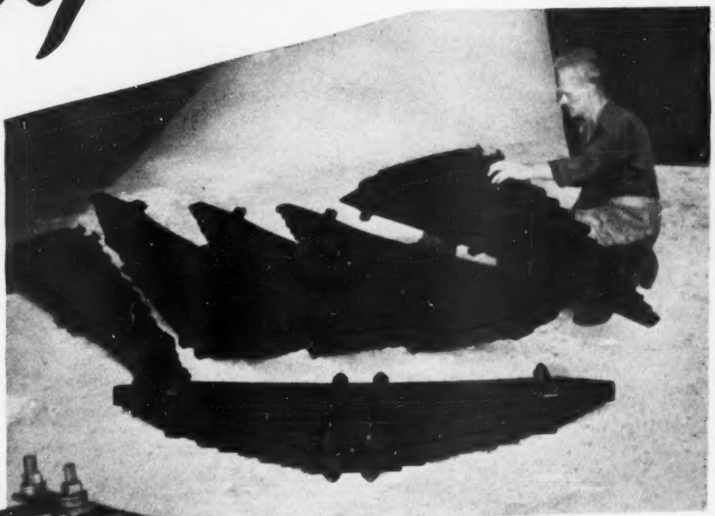

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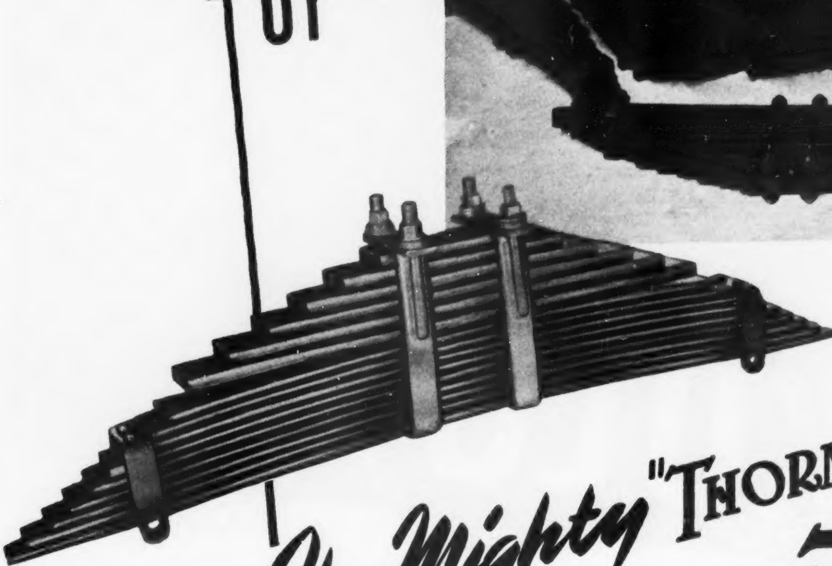

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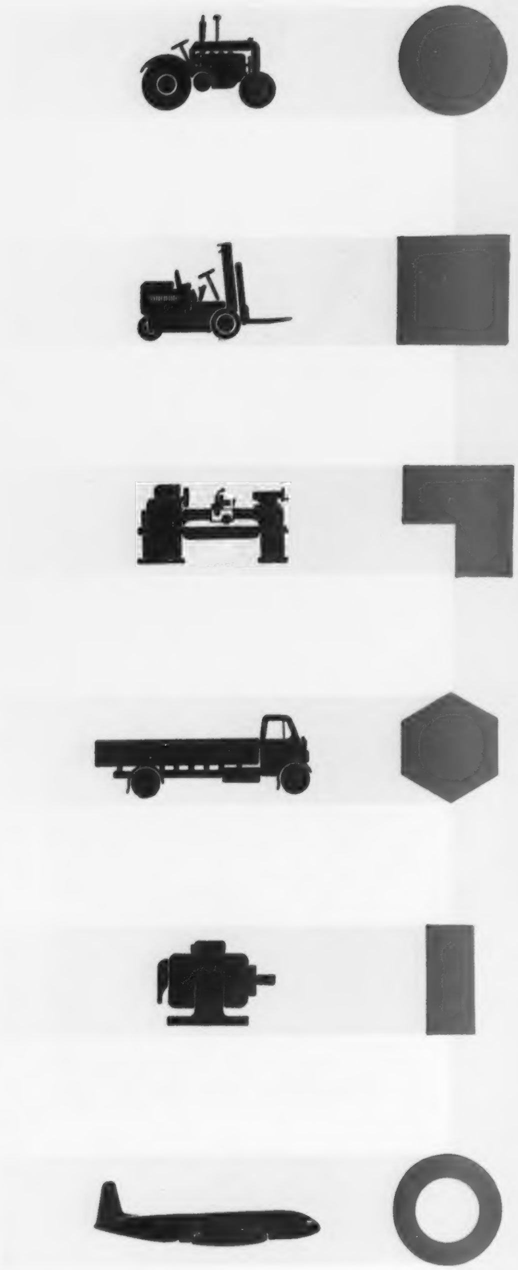

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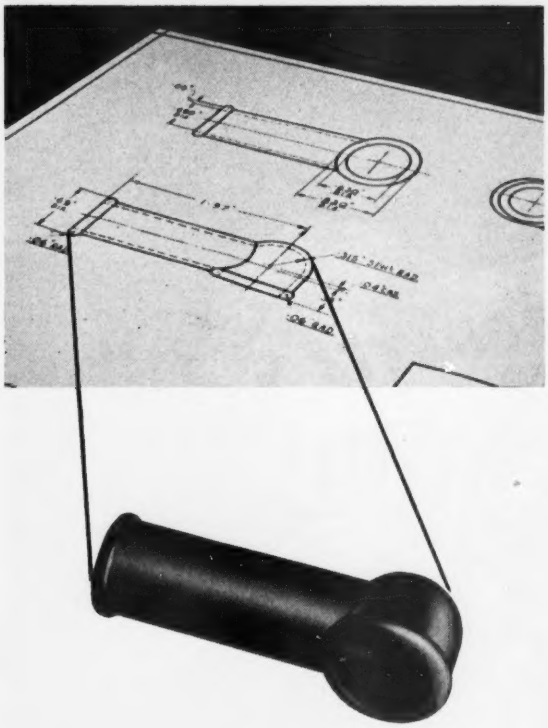

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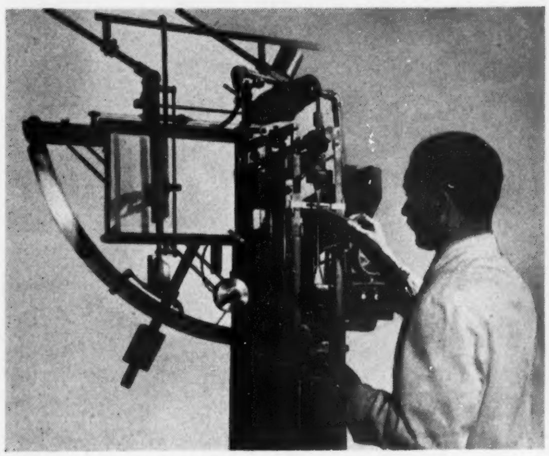

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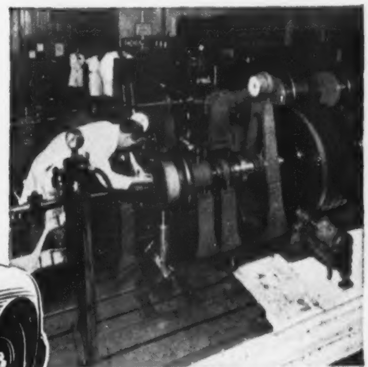

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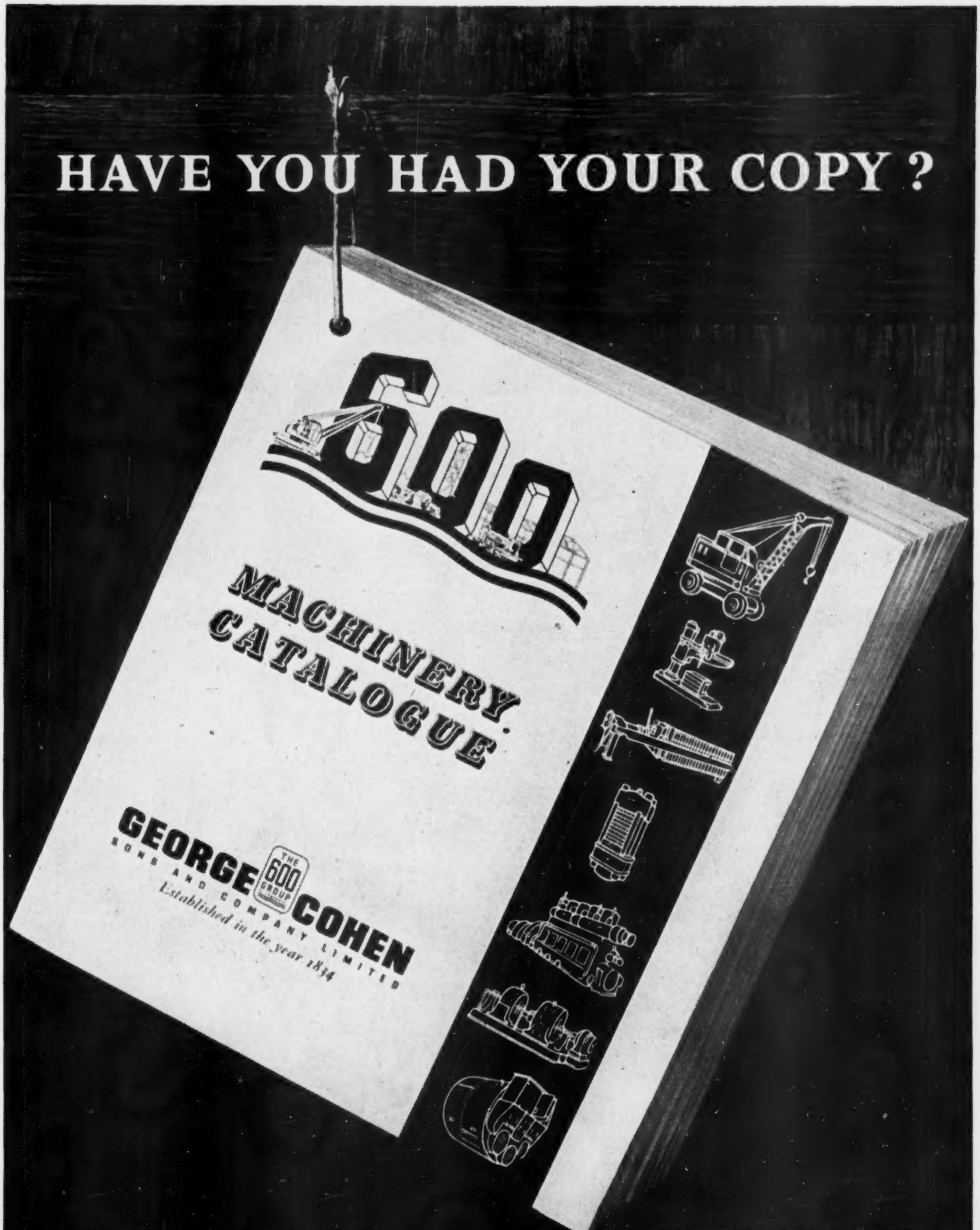
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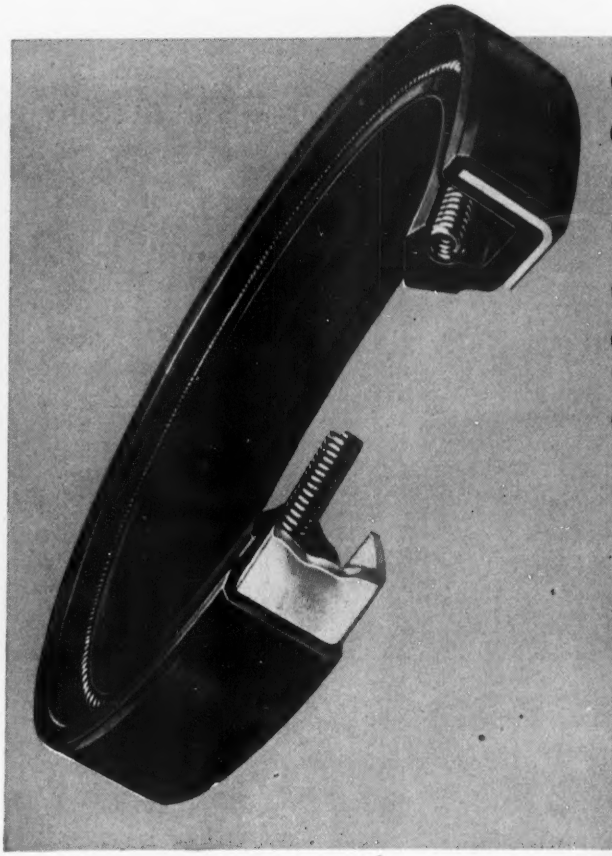

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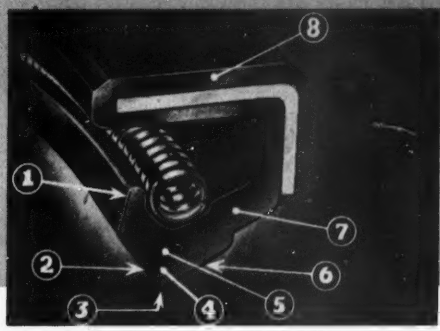

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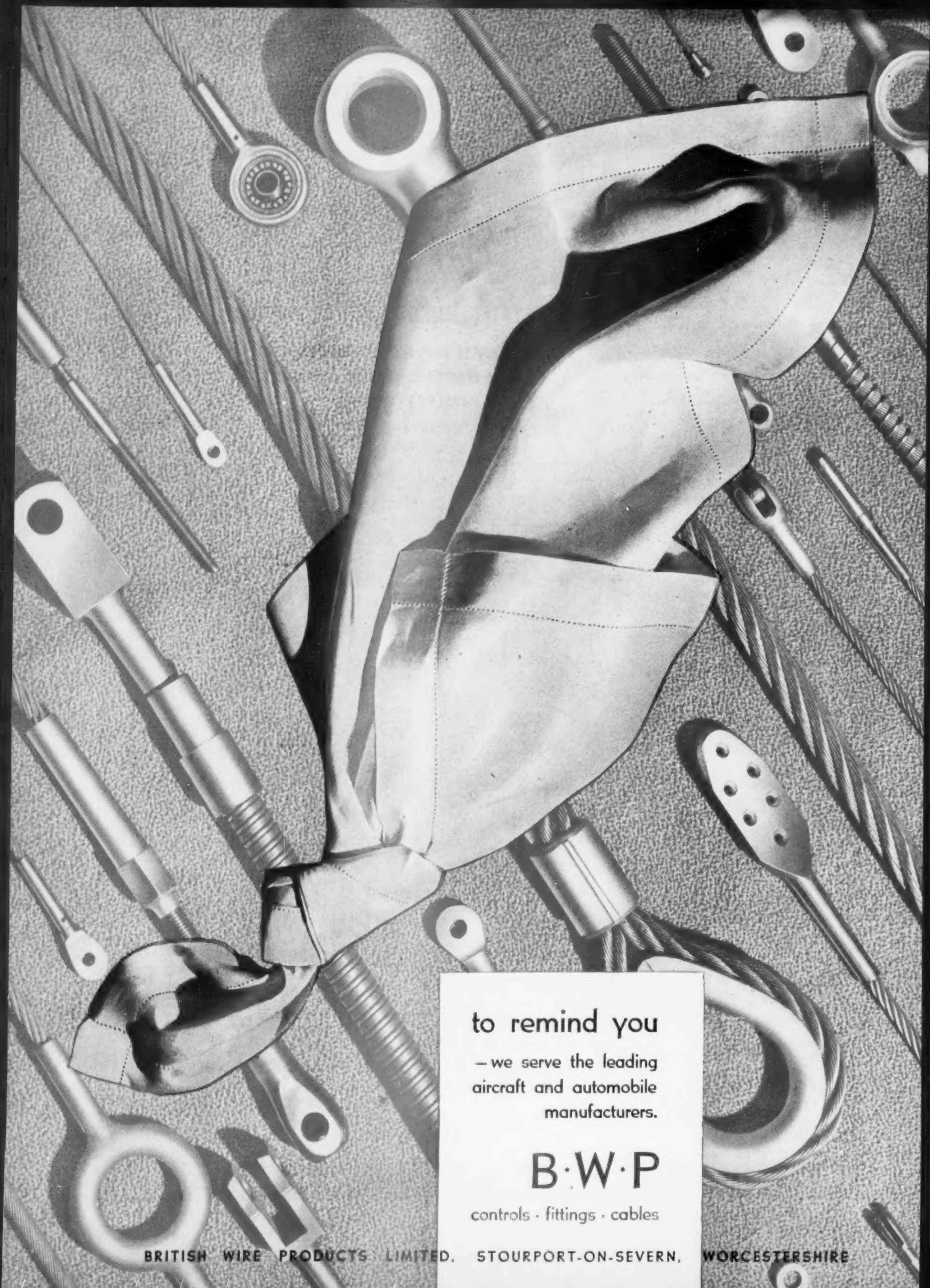
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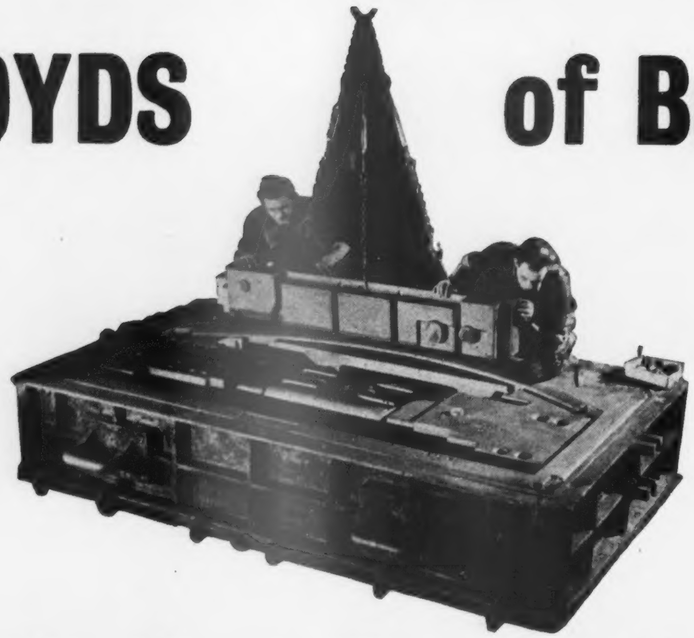

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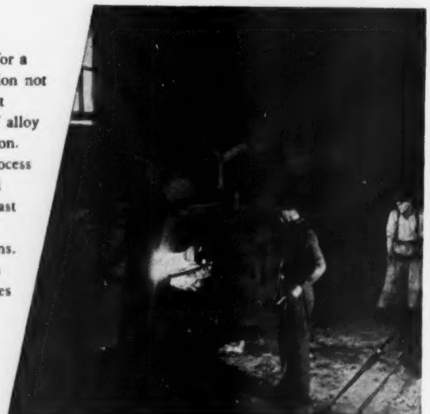

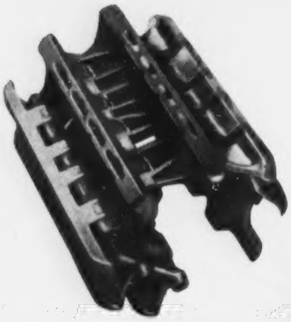

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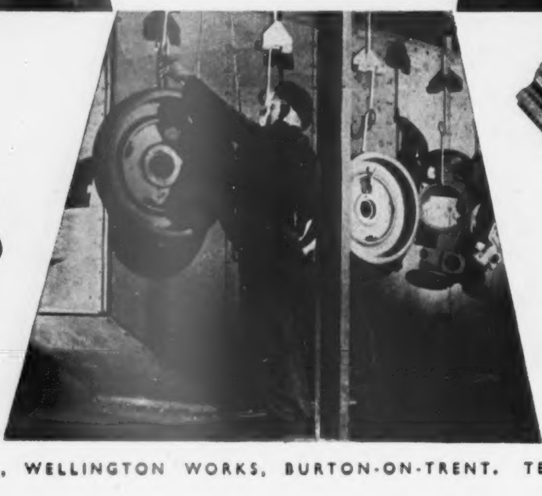

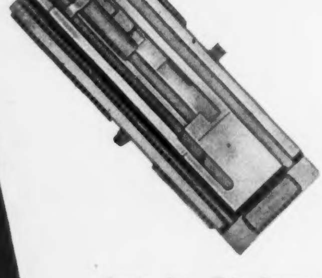

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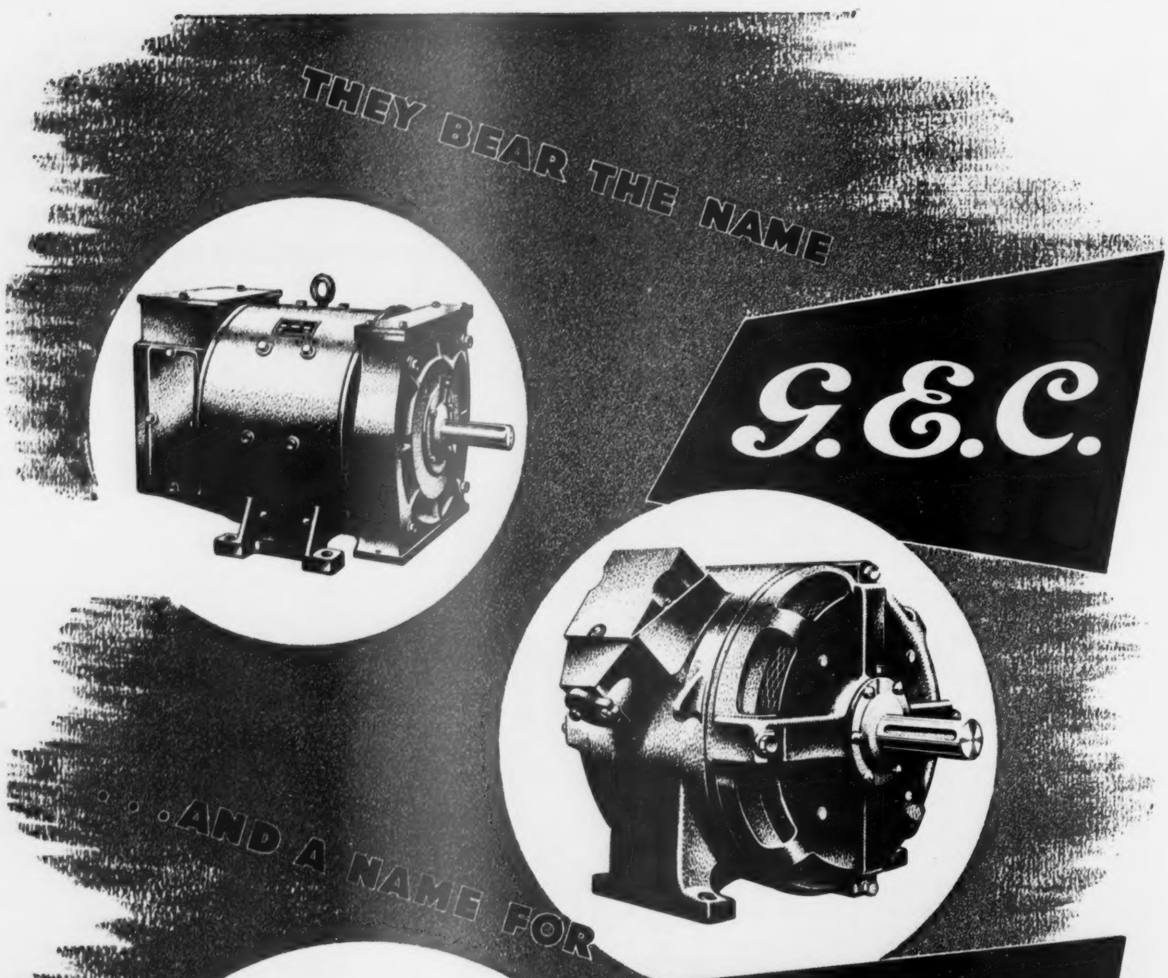
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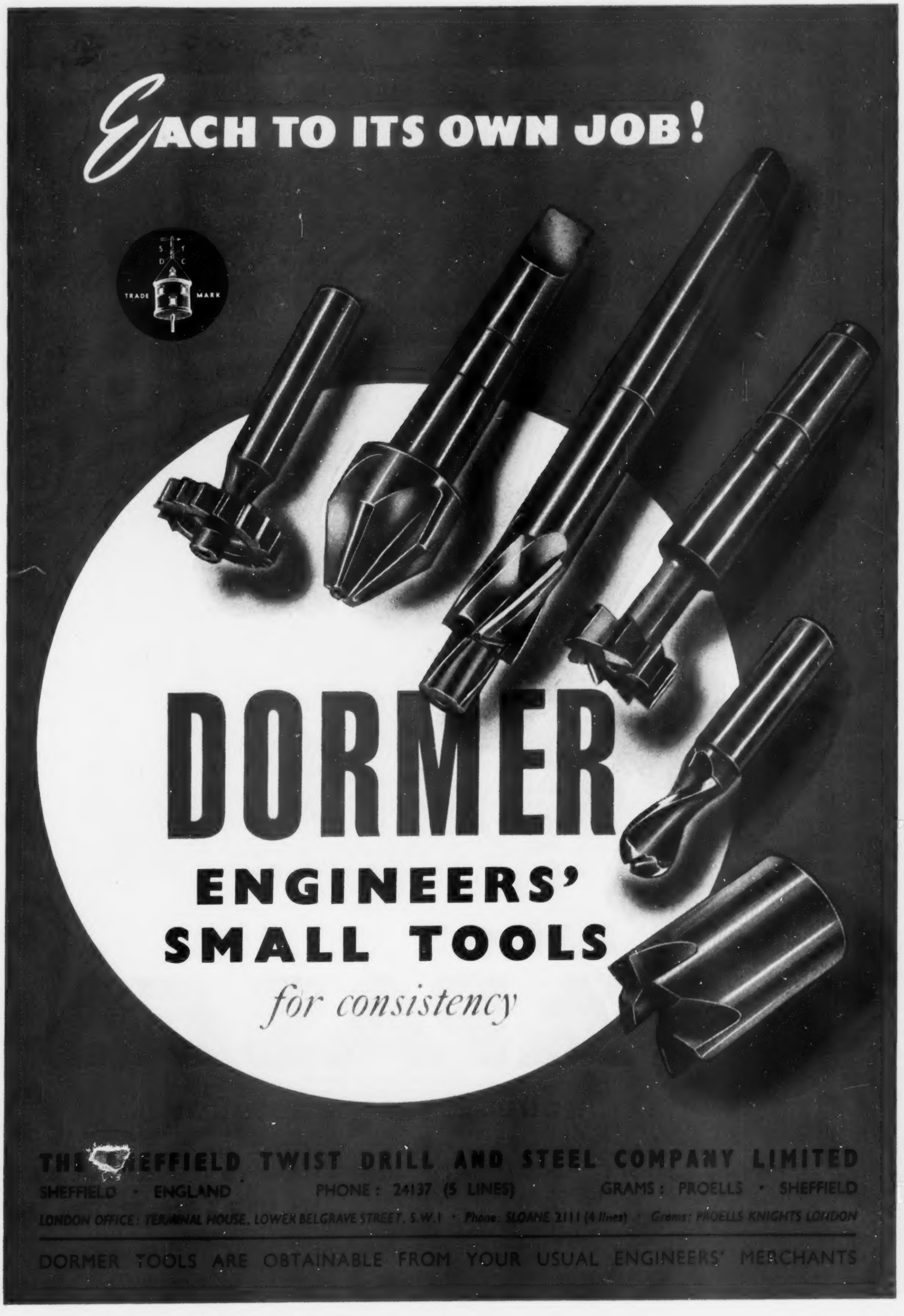
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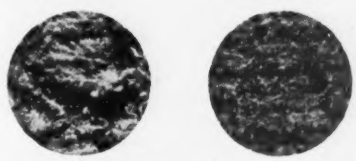

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Design, Materials, Production Methods, and Works Equipment

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VOL. XLIII No. 570 SEPTEMBER, 1953 PRICE 3s. 6D.

Body Insulation

BECAUSE we have a temperate climate in this country, there is a tendency to overlook some of the difficulties that face overseas motorists. From time to time manufacturers concentrate for a while on such problems as thermal insulation of motor bodies; careful investigations are instituted and decisions are taken as to what design features are necessary. After this, it is a case of out of sight, out of mind; no further progress is made and, as a result, there is a tendency to fall behind American competitors who have the advantage that this problem is always with them in their own country. That the Americans have paid particular attention to this feature is demonstrated by the fact that a number of their cars have glass fibre mat stuck to roof panels and other components.

Even in the British Isles, a car, except when it is in the shade, can be almost unbearably hot to enter on sunny days. So effective insulation, if provided at an economical price, would undoubtedly have sales appeal. Moreover, such insulation would also serve to keep heat inside the body during cold weather, and alleviate the difficulties involved in providing a heater installation large enough to maintain a reasonable interior temperature. That the problem is not entirely insoluble is demonstrated by the fact that after a caravan and the car that tows it have been standing together under the same conditions for a number of hours, the caravan is usually much cooler inside than the car.

Differing requirements

However, it must be admitted that the design requirements are considerably different as between a car and a caravan. In a car, space is at a premium, whereas an inch thickness of insulation, whether it be simply air space between double walls, or a filling of proprietary material, can be accommodated relatively easily in the larger vehicle. The curved contours of car body panels make it difficult to fit insulating material. This disadvantage is the more marked because many of the best insulating materials are supplied in relatively stiff sheets. Moreover, double skinning, which is so easy on the flat panels of caravans, is generally difficult and expensive on cars. In addition, it would inevitably increase the weight of the vehicle, thus having an adverse effect on both performance and fuel consumption.

Probably one of the most important differences between the two types of vehicle is that cars are almost invariably required to travel many times the mileage of caravans. This means that materials used for their insulation must have strength enough to hold together for very long periods of running under conditions of continual vibration. Moreover, if the insulation is glued to the panel, there must be no tendency for the surface layer to separate from the main body of the material.

This narrows the choice of insulating materials appreciably; in fact, it eliminates all but the more expensive. The best insulators are generally those whose density is low, and in which there is a relatively large amount of air space, usually between fibres or in pores. These materials need to be quilted, preferably between two layers of cloth or paper, in order that they may fulfil the strength requirements already mentioned.

Current practice

It would be far from the truth to suggest that nothing is done at present to insulate modern cars. However, the measures that are taken are generally aimed at deriving the greatest amount of benefit at the lowest possible cost. This is a very sound principle, and we would not suggest that any other should be adopted. The point at issue is whether or not the current measures are carried far enough. We are of the opinion that more attention should be paid to roof panels, but before elaborating on this theme we shall survey what is already done with regard to insulation and then develop the argument for carrying it a stage further.

Underfelt is at present used with carpets. In some cases it is as much as ⅜ in thick; but, generally it is about ¼ in thick, and it invariably treads down in use to something like half its original thickness. The felt and carpet perform four distinct functions. The use of both together improves the wear resistance of the floor covering. They give a more pleasing interior appearance and improve the comfort of the vehicle by insulating passengers' feet from vibration. The acoustical absorption properties of the body are improved to such an extent that the difference between a covered and uncovered floor may be likened to that between a furnished and an unfurnished room in a house. It also gives a large measure of thermal insulation, and to a lesser degree acoustical insulation. There can be little doubt that good value is obtained for the small expenditure on underfelt.

Materials such as felt and glass fibre mat are often clamped against dash panels and held there by means of

millboard facing. One object of doing this is to insulate the body from engine noise, although materials of this type are not ideal from the point of view of acoustical insulation. The other object is to prevent the transfer of heat from the engine compartment to the body. Since the absorbent material is usually covered with millboard, it is unlikely to have much effect on the reverberation characteristics in the car.

Apart from these features there are others which, although not intended primarily to provide thermal insulation, do have a beneficial effect in this respect. One of these is the inner panel on the doors. The main function of these panels is to hide the window lift mechanism and present a clean interior finish between the door pillars. The seat squab generally provides reasonably good insulation at the rear, so the only parts remaining to be treated are the areas of glass and the roof.

A number of cars in the luxury class are double glazed, but this feature is incorporated primarily to prevent the misting of the windows in cold weather. However, double glazing is not likely to become popular, because it is both heavy and costly and is unlikely to prevent the passage of the radiant heat of the sun's rays into the car. In any case, the thermal conductivity of glass is not very high, so that little heat is likely to be conducted through the glass away from the interior of the car.

Roof panels

From the foregoing, it follows that in most cases the roof panel is the only one that is not treated in a manner likely to provide some effective thermal insulation. Moreover, it is this part of the vehicle which is most exposed to the sun. Therefore, if further progress is to be made, it is to this panel that attention should be turned.

The problems to be overcome here are different from those associated with the insulation of other parts of the body. The insulating material must either go between the head-lining and the panel, or it must be integral with head-lining. If it goes between the two, it must be a firmly bonded material so that, no matter what the temperature or humidity of the climate in which it is used, it will not disintegrate and fall on the head-lining. This is especially necessary when the insulating material is stuck to the panel.

In a number of American made cars, the material used to line roof panels is fine glass fibre mat bonded with resin. This material is exceptionally light, so there is little tendency for it to fall down or to disintegrate under its own weight. Moreover, the glass fibre does not deteriorate or absorb moisture, and it is vermin-proof. Whether or not such an arrangement is suitable for cars used in Britain is open to doubt, since it may not be durable enough for our vehicles.

Insulating materials

There are two effects that must be taken into consideration where long life is required. One is the ageing of the adhesive, and the other is corrosion of the metal surface to which the glass fibre mat is stuck. The resistance to ageing of the materials used, and the effectiveness of their adhesion to metal surfaces, can only be determined satisfactorily by practical tests over relatively long periods of time in a variety of climates.

Insulating materials in the form of a relatively stiff felt, or soft cardboard are available. They are generally made of kapok, and are sometimes reinforced by a layer of kraft paper stuck on one or both sides. This type of material can be held up against the roof panel by the listing strips used for trimming purposes, and no adhesive is needed. The principal disadvantages of these materials are that many of them break easily during handling, and it is difficult to fit them into a domed roof panel.

The roof panels of some of the less expensive cars, and a number of truck cabs are trimmed with millboard attractively finished on one side. Under these conditions it is relatively easy to apply the softer insulating materials by fitting them between the board and the panel. In fact, this is done on some truck cabs. If insulation is considered necessary for these vehicles, surely it is a feature that is also desirable for private cars. However, millboard is not ideal for trimming because, not only does it reflect sound waves and cause reverberation, but it is also liable to resonate and in that way increase the noise level in the car.

One of the simplest, yet often ignored, ways of reducing the rate of increase of interior temperatures in a car parked in the sun is to finish the exterior in a light colour. White is, of course, the best from the point of view of reflecting radiated heat from the sun, but it is a somewhat impracticable because of the difficulty of keeping it clean. However, pale pastel shades are suitable and are becoming increasingly popular. A highly polished finish is also a marked advantage. There is little doubt that a great deal of good could be done by sales departments in educating the public on this subject.

CONTENTS

THE FORD COST CUTTER ENGINE

The New Unit Designed for the 2 and 3 ton Thames Trucks

A NEW four cylinder petrol engine, the Cost Cutter, has been introduced by the Ford Motor Co. Ltd., to take the place of the V8 engine hitherto fitted to the 2 and 3 ton Thames models as well as to the larger trucks. The V8 power unit, or as an alternative the Perkins P6 diesel engine, is still used in the bigger vehicles. The makers claim that they are the only manufacturers to design and produce in large quantities an engine specifically for the 2 and 3 ton range.

When the design was first contemplated, the principal aims were to produce an engine of appropriate power which was at the same time economical to run and maintain. In addition, considerable pains were taken to incorporate features that would give it a long and trouble-free life. For instance, a five-bearing crankshaft of generous proportions and separate cylinder liners cast from an iron with a high chromium content have been employed. With this alloy of cast iron, it is not necessary to use chromium plated compression rings.

From the point of view of economy, it was considered that the four-cylinder layout was preferable to the V8 arrangement used in the bigger engines. Moreover, many of the components, including the crankcase but excluding the cylinder head, are common both to this engine and to the unit for the Fordson Major tractor described in the April 1952 issue of *Automobile Engineer*. Not only does this make it possible to supply spare parts at a more economical price, but, because a smaller number of different spares need be kept in stock, it is more likely that components required can be taken straight off the shelf and the operator will not suffer the inconvenience of having to wait for delivery.

SPECIFICATION

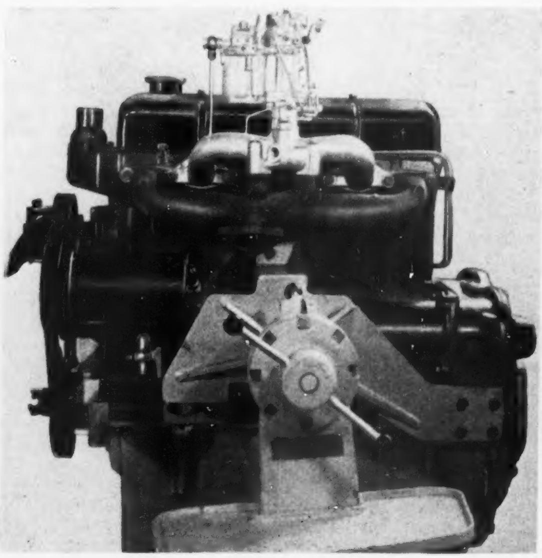
Four cylinders. Bore and stroke 100 mm × 115 mm. Swept volume 3,611 cm³. Maximum b.h.p. 70 at 2,800 r.p.m. Maximum b.m.e.p. and torque respectively 113 lb/in² and 165 lb-ft at 1,500 r.p.m. Compression ratio 6·0:1. Fully balanced, five bearing, cast alloy steel crankshaft. Push rod operated overhead valves. Zenith 36 VIS downdraught carburettor with 32 mm choke.

The engine has a swept volume of 3,611 c.c. and develops 70 b.h.p. at 2,800 r.p.m. Its bore : stroke ratio is 0·87 : 1. This ratio has been determined largely by the length of the bearings needed to carry the five main journals. Because a cast crankshaft is used, the journal diameters are fairly large so that only relatively short bearings are necessary to limit the maximum bearing pressures to a reasonable figure. Accordingly, from the point of view of making the engine as short as possible, it was undesirable to incorporate cylinders of a larger bore with a correspondingly shorter stroke. The connecting rod length : stroke ratio is 1·77 : 1.

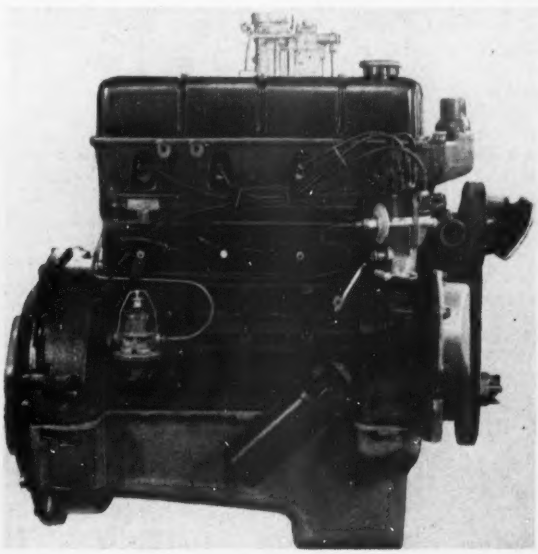
At 2,800 r.p.m. the mean piston speed is 2,114 ft/min. The maximum b.m.e.p. is 113 lb/in² and the maximum torque is 165 lb-ft, both values being obtained at 1,500 r.p.m. Without the clutch, the dry weight of the engine is 605 lb, so that the b.h.p. developed per pound is 0·116. In terms of b.h.p/in² piston area, a figure of 1·437 is obtained, and the b.h.p. developed per litre is 19·4. The overall dimensions without the air filter fitted are: height 35½ in; width 22½ in; length, including flywheel, 31½ in.

Silentbloc engine mountings are employed. At the front, the engine plate is extended on each side and the extensions are turned back on themselves to form vertical tubular housings for the holding-down bolts. These bolts are inserted from above and the threaded lower end of each extends through the centre of a Frustacon unit bolted to a bracket on the frame. A self-locking nut is screwed on to hold the assembly together.

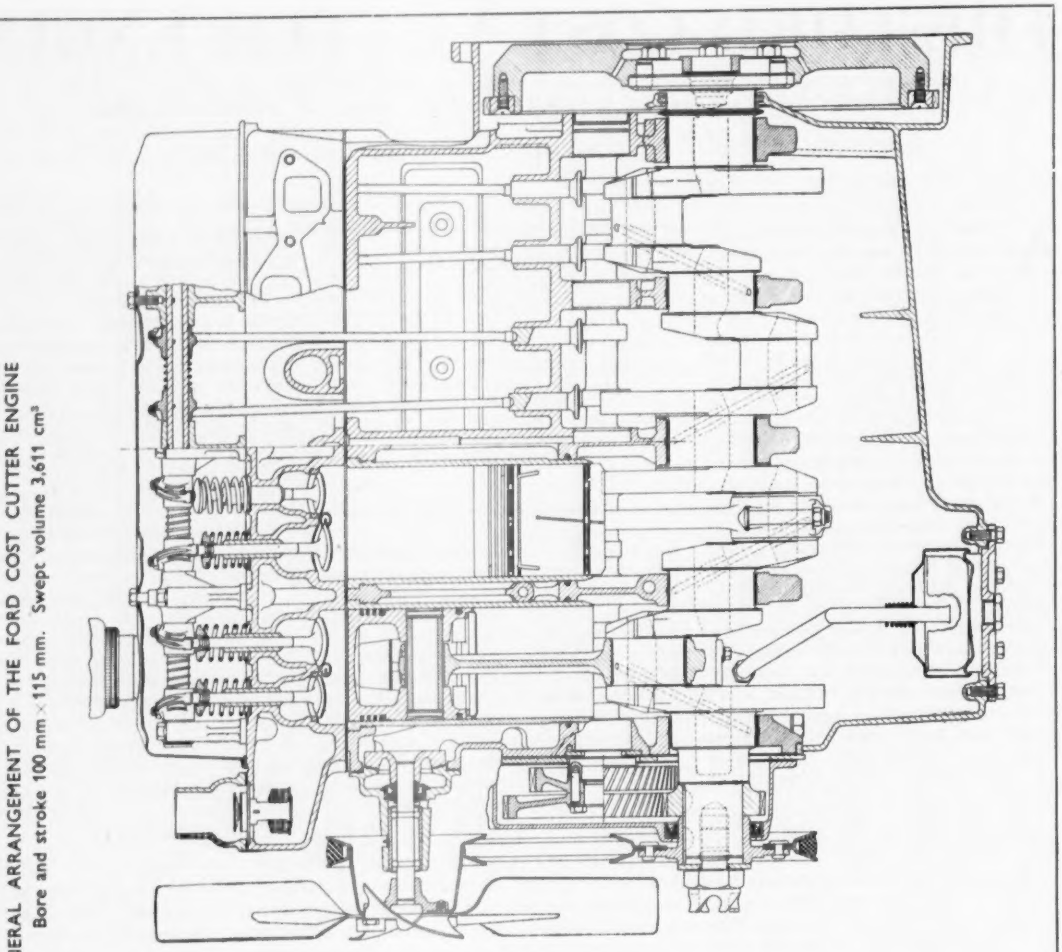
At the rear, the mounting is similar to the one used on the V8 unit. It consists of a large diameter rubber ring with a deep groove round its periphery; a plate engages in the groove. The rubber ring and plate are carried in a channel formed between the flange round a dished ring and the rear face of the gearbox to which the dished ring



A view from the left-hand side of a Ford Cost Cutter engine mounted on a servicing stand



The fuel pump is on the right-hand side where it is well away from the influence of heat radiated from the exhaust



GENERAL ARRANGEMENT OF THE FORD COST CUTTER ENGINE
Bore and stroke 100 mm × 115 mm. Swept volume 3,611 cm³

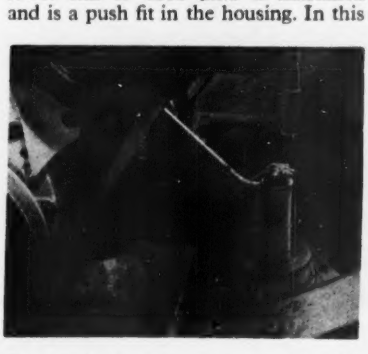
is bolted. Bolts through each side of the plate secure it to brackets on the frame.

Crankcase, cylinder block and liners

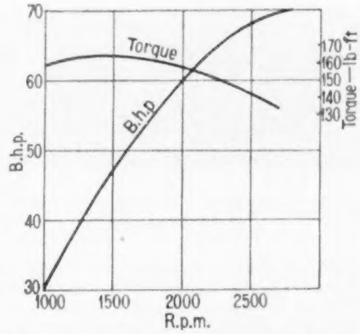
The cylinder block and crankcase unit is made of cast iron having a B.H.N. of 179-228, and it is used in conjunction with a cast aluminium sump. At the rear, extensions of both the sump and the crankcase form the flywheel housing, the end face of which is machined to carry the bellhousing. A bonded asbestos-yarn ring forms the seal round the tail end of the crankshaft. This ring is split diametrically and housed in channels formed in the rear wall of the sump and in a separate aluminium die-casting bolted to the rear of the crankcase.

The front end of the crankcase and the front journal bearing cap are machined to carry the timing cover which is bolted on together with the 5/32 in thick front plate and two Langite 132 joint washers. Above this is an aperture in the block, which is machined and faced to take the spigoted-in water pump unit. Below the timing cover, the front oil seal is formed by a rectangular section Neo Langite strip in a semi-circular housing machined in the periphery of the front face of the front journal bearing cap. The sump bears against the cork which is held in position by the front plate.

Sheepbridge Stokes, centrifugally-cast, iron cylinder liners are fitted. Their wall thickness is 3/16 in and the minimum jacket space between the liners is 11/16 in. The bores are honed to 15 r.m.s. At the top end a flange, 7/16 in deep by 1/8 in wide, seats on a shoulder machined in the block. This flange is held down by the cylinder head and gasket. The flexibility of the gasket permits a certain amount of machining tolerance to be allowed on the flange dimensions and the shoulder depth. An undercut is machined in the liner immediately below the flange to avoid the necessity of having continually to re-dress the edge of the grinding wheel during manufacture. Radial location at the top end is effected by an integral collar below this recess, which is a push fit in the housing. The lower end of each liner is machined and is a push fit in the housing. In this



One of the front engine mounting units



Performance curves of engine without fan but with generator working

housing, there is an annular groove that houses a rubber sealing ring.

The crankcase walls and the three webs for the intermediate journal bearings extend down to about 1/8 in below the level of the axis of the bearings. Each of the cast iron bearing caps is shouldered to register in recesses which are cut to a depth of 1/8 in in the webs, to furnish lateral location. They are held down by two 1/2 in diameter En 111A set bolts locked by heavy tab washers.

Crankshaft, connecting rod and piston assembly

A cast steel crankshaft is employed, the material specification being: C 1·2-1·45 per cent, Mn 0·7-0·9 per cent, Cu 1·5-2·0 per cent, Si 0·85-1·1 per cent, Cr 0·4-0·5 per cent, P 0·1 per cent maximum and S 0·08 per cent maximum. There are five main journals and axial location is effected at the centre one. There, Vandervell, steel-backed white metal, split thrust washers in recesses on each side of the cap and housing bear against the inner face of the two adjacent crankshaft webs. The thrust washers are located against rotation by a tongue in each lower half, which extends vertically downwards and engages in a slot machined in the end face of the bearing cap. All the crankpins and journals are cored to reduce the weight of the shaft.

PISTON RING DATA

	Compression Rings	Oil Control Rings
Gap	0·011 -0·016 in	0·011 -0·016 in
Side clearance	0·0014-0·0034 in	0·0015-0·0035 in
Face width	0·0928-0·0938 in	0·1865-0·1875 in
Radial thickness	0·151 -0·158 in	0·151 -0·158 in
Depth of groove in piston	0·168 -0·1735 in	0·1655-0·173 in

Steel backed, indium plated copper-lead bearings are employed to carry the 3 in diameter journals. The effective bearing lengths are: front and intermediates 0·905 in, centre and rear 1·28 in. Their diametral clearance is 0·001-0·0028 in. Location is effected in the usual manner by pressed-out tags on the shells, engaging in slots machined in their housings.

The overall length of the crankshaft from the front of the front web to the back of the rear web is 19⅛ in. There is a certain amount of overlap of the 2½ in diameter crank pins and the journals, and the crank webs are 1 in thick by 4⅛ in wide. Towards the rear end, an oil thrower ring is formed on the shaft immediately in front of the oil seal. At the extreme rear there is a 5⅞ in diameter by ½ in thick flange on which the flywheel is spigoted.

A Ford specification cast iron, similar to B.S. STA 8, is used for the flywheel which is secured by six tab washered ½ in diameter set bolts, and located by two ⅜ in diameter dowels. The overall diameter with the ring gear fitted is 16¼ in, and 15½ in without, and the overall thickness is 2½ in. Its weight, including the ring gear, is 63 lb. In the centre of the rear face of the flywheel is a circular lipped recess. Holes are drilled from inside the lip to drain to the front of the flywheel any lubricant which might be flung out by centrifugal force from the ball bearing which supports the front end of the clutch shaft. This prevents the lubricant from getting on to the clutch faces.

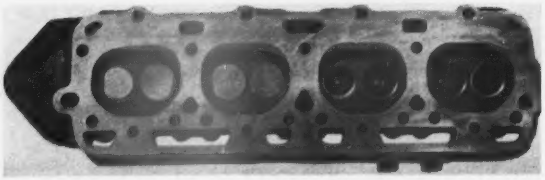
Six countersunk set screws are used to secure the ring gear on the shouldered periphery of the flywheel. There are 128 teeth on the gear, which is made of a steel containing C 0·46-0·53 per cent, Mn 0·75-0·95 per cent. On overhaul, the ring may be turned through multiples of 60 deg relative to the flywheel so that wear may be distributed more evenly on the teeth.

The forged H-section connecting rods of En 18C have a centre-to-centre length of 8 in. The cross sectional dimensions of the rod are 11/16 in wide over the flanges by 1 in deep, and the web thickness is 7/32 in. Two En 18B, 7/16 in diameter bolts retain the bearing caps on the big ends which are split at 90 deg to the axis of the rod. There would be no object in splitting them at any other angle, since the big ends are only 3¾ in wide and may be withdrawn through the cylinder bores. A flat on the head of each bolt registers against the shoulder on the rod and self-locking nuts are fitted. The big end effective bearing length is 1·345 in and the diametral clearance 0·001-0·0028 in.

A steel backed, lead bronze bush of

Arrangement of the rear engine mounting

1·39 in effective length is pressed into the small end and is fed with oil splashed into two countersunk $\frac{3}{32}$ in drillings in the top of the rod. An En 206 gudgeon pin is employed and the diametral clearance in the bush is 0·0001-0·0007 in. The outside diameter of the pin is $1\frac{1}{4}$ in while its inside diameter is about $\frac{7}{8}$ in. The bearing length at each end of the gudgeon pin in the piston boss is $\frac{13}{16}$ in, and the diametral clearance is from 0-0·0002 in. Location is effected in the conventional way by means of circlips in grooves in the piston bosses.



The thermostat is housed in a forward extension of the cylinder head casting

Castings for the pistons are supplied by Aeroplane and Motor Aluminium Castings Ltd. and by Northern Aluminium Company Ltd. They are machined and barrel ground by Fords. They are of LM 13 WP aluminium alloy and are tin plated to assist running-in. The skirts are split on the side opposite to the thrust face and a thermal slot is incorporated on each side below the top oil control ring. Two oil control rings are fitted, one above the gudgeon pin and the other below it, and three compression rings are used. All the compression rings are supplied by Wellworthy Piston Rings Ltd. or Hepworth and Grandage Ltd., and their principal dimensions, together with those of the oil control rings, supplied by Hepworth and Grandage, are shown in the Table. The top compression ring is square-faced and hardened and tempered, while the second and third ones are conventional taper-faced rings.

Timing gear, valve gear and camshaft

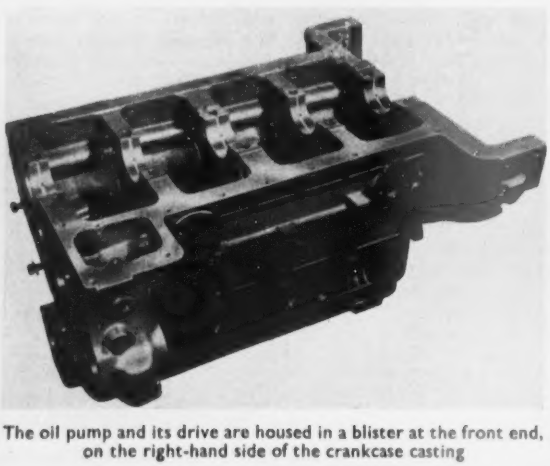
Helical gears of cast iron are employed to transmit the drive from the crankshaft to the camshaft. An additional gear, mounted on a separate spindle to the right of the camshaft and driven by a gear behind the half speed wheel, takes the drive for the oil pump and distributor. On all gears, the helix angle is 25 deg. The driving gear, together with the cast iron boss of the pulley, is carried on the $2\frac{1}{32}$ in diameter front end of the crankshaft and both are driven through a Woodruff key. They are pulled up against a shoulder on the crankshaft by a special bolt, which incorporates the dogs for the starter handle, and a $\frac{3}{16}$ in thick mild steel washer. The overall length of the crankshaft front extension is $3\frac{7}{8}$ in and the diameter of the portion between the helical gear and the front journal bearing is $2\frac{1}{2}$ in. A lip type oil seal supplied by George Angus and Co. Ltd. or Super Oil Seals and Gaskets Ltd. is housed in the LM 20 M cast aluminium timing cover and bears on the boss of the pulley. The pulley rim is a two-piece steel pressing and it is riveted to a flange around the boss.

VALVE DATA

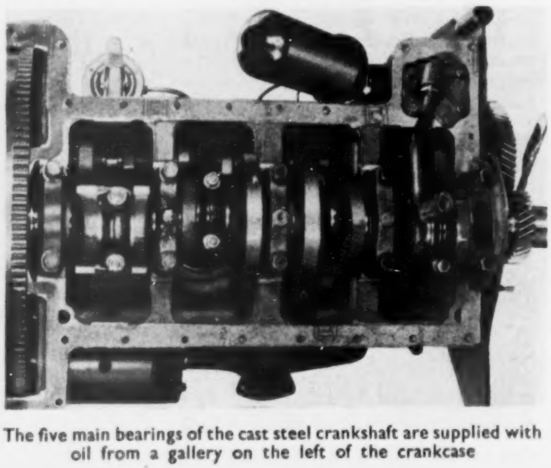
	Inlet	Exhaust
Material	S.A.E. 8645	B.S. 970, En 59
Head diameter	1·720 –1·730 in	1·533 –1·543 in
Throat diameter	1·560 –1·565 in	1·372 –1·377 in
Stem diameter	0·3731–0·3742 in	0·3723–0·3733 in
Valve seat material	Plain	Cast Tungsten alloy
Valve seat angle	30 deg	
Spring material	D.T.D. 5A, cadmium or zinc coated	
Spring load (fitted)	160–172 lb	
Spring length free	2·48 in	
Spring length installed	1·98 in	
Number of coils	8·8 total, 7 free	
Coil diameter	0·994 –1·006 in I.D.	
Wire gauge	0·1605–0·1665 in	
Valve lift	0·428 in	
Rocker ratio	1·452 : 1	
Valve guide material	cast iron	
Valve guide inside diam	0·375 –0·376 in	
Valve guide outside diam	0·6260–0·6265 in	
Valve guide length	2·22 in	
Tappet clearance	0·015 in (hot)	
Valve opens	13 deg B.T.D.C.	49 deg 49 min 12 sec B.B.D.C.
Valve closes	48 deg 49 min 12 sec A.B.D.C.	12 deg A.T.D.C.
Ignition timing	6 deg B.T.D.C. (advanced)	

The camshaft is made of an alloy cast iron containing 0·25 per cent Cr and 2·5-3·0 per cent Cu. Its diameter is 1·12 in. A diametral pitch of 10 has been adopted for the half speed wheel which, together with the gear engaging the oil pump and distributor drive, is mounted on the $1\frac{5}{8}$ in diameter front end of the camshaft, behind which is a flange to which the gears are secured by three $\frac{5}{16}$ in diameter set bolts. To ensure the correct timing relationship between the settings of the two gears and the camshaft, a $\frac{3}{16}$ in diameter dowel is pressed into a blind hole in the camshaft and engages in a hole through the two gears. The front end is covered by the tab ring which is used to lock all three bolts and prevent the dowel from coming out.

Immediately behind the flange is a groove for the $\frac{3}{16}$ in thick sintered thrust plate. This plate is circular in shape and is divided so that it may be assembled into the groove. It is housed in a recess machined in the front wall of the crankcase and is located against rotation by a peg in the upper portion. This peg registers in a hole drilled in



The oil pump and its drive are housed in a blister at the front end, on the right-hand side of the crankcase casting



The five main bearings of the cast steel crankshaft are supplied with oil from a gallery on the left of the crankcase

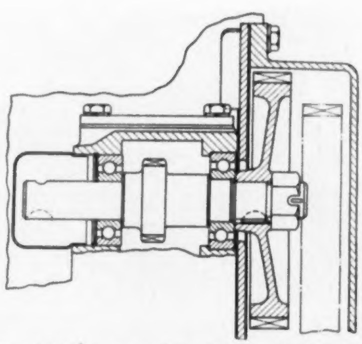
the crankcase wall. The two parts of the thrust plate are retained in the recess by the front plate.

All five camshaft bearings are carried directly in bosses in the crankcase walls and webs. They are 2·0595-2·0600 in diameter. The front bearing journal is 1·03 in long, the intermediate ones are 0·88 in long and the length of the rear one is 0·82 in. Between the cams, the shaft diameter is $1\frac{1}{8}$ in. This relatively large diameter, and the support furnished by the bearings, which are spaced only $4\frac{1}{16}$ - $4\frac{5}{16}$ in apart, undoubtedly contribute towards the efficiency of the valve operation.

Mushroom type tappets of case hardened En 207 steel are employed. They are $\frac{5}{8}$ in diameter and their radial clearance in the guides is 0·0015-0·003 in. The guides are carried in $1\frac{5}{8}$ in long bosses in the crankcase. A rotary motion is imparted to the tappets by the offset, about $\frac{1}{8}$ in, of their axes from the cam centres. En 8C push rods are employed. They are $\frac{5}{16}$ in diameter and their effective length is 11·9 in. Each has a ball end up-set at the bottom, where it bears in a hemispherical seat in the end of the tappet, and a cup end at the top. Both ends are cyanide hardened.

The malleable cast iron rockers have chilled bores and end pads. At the push rod ends, En 1A steel tappet adjusting screws are fitted. They are secured by lock nuts and their ball ends are cyanide hardened. An En 43B shaft is employed. It is induction hardened locally where the rockers bear. The shaft is about $\frac{3}{4}$ in outside diameter and $\frac{15}{32}$ in inside diameter. Each pair of rockers is held apart by the conventional arrangement of a compression spring around the shaft, but spring seating washers are not used.

Five cast iron pedestals support the shaft and each is secured by one $\frac{5}{16}$ in diameter set bolt. On assembly, the rockers, springs and pedestals are threaded on to the shaft and retained by mushroom type, mild steel end plugs which are secured by pins through the shaft. These pins are prevented from falling out by the end pedestals which, when the assembly is complete, cover the holes in which the pins are carried. The shaft is located by means of a dowel ended set bolt screwed into number 3 pedestal and engaging in a hole in the shaft. Bosses on top of numbers 2 and 4 pedestals are drilled and tapped for the $\frac{5}{16}$ in diameter set bolts that hold down the rocker cover. Fibre seating washers are employed under the heads of the bolts, and the seal at the joint between the pressed steel rocker cover and the cylinder head is formed by a Langite joint washer.



A spiral gear on a short horizontal spindle engages another on the vertical drive for the oil pump and distributor

CAMSHAFT PERFORMANCE DATA AT 2,800 R.P.M.

Maximum tappet positive acceleration (flank)	4,300 ft/sec²
Maximum tappet negative acceleration (nose)	1,520 ft/sec²
Maximum tappet velocity	6·05 ft/sec
Lift at cam	0·305 in
Nominal period of cam	120 deg 54 min 36 sec

Details of the valves and springs are given in the Table. The valves are carried in line in the cylinder head at an angle of 24 deg from the vertical. Plain cylindrical guides are fitted, and the cadmium coated valve springs seat directly on the head. At the top end of the spring the usual retaining washer and split collet arrangement has been adopted. Rubber rings are fitted in grooves round both the inlet and exhaust valve stems to form a seal inside the lower end of the collets and prevent an excessive amount of oil from running down into the guides.

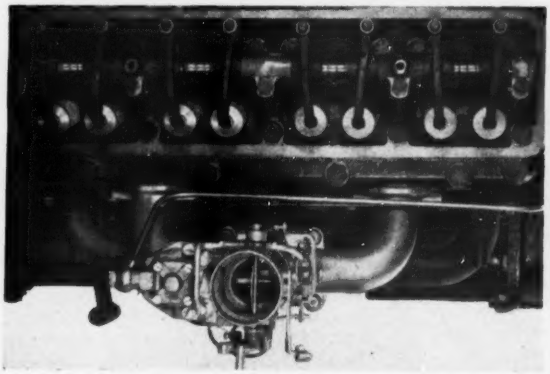
Cylinder head, manifolds and carburettor

Wedge shaped combustion chambers are incorporated in the cast iron cylinder head. The machined lower face of the head extends over the cylinder bore for a distance of about $\frac{1}{4}$ in to give a squish effect as the piston comes up to top dead centre. The sparking plugs are carried in bosses recessed in the right-hand side.

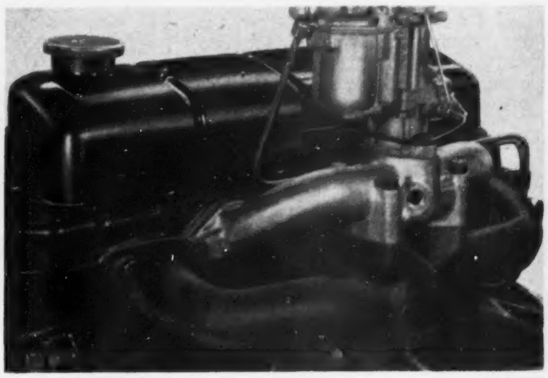
Seventeen $\frac{1}{2}$ in diameter En 111A set bolts hold down the cylinder head. These bolts are screwed into tapped holes in bosses at the top of the block. The heads of those on the right-hand side are outside the rocker cover. A gas-tight seal at the cylinder head joint is effected with a copper and asbestos gasket. The overall depth of the casting is $3\frac{5}{8}$ in and its width is 7 in.

Separate exhaust ports are employed and a cast iron manifold is secured by seven $\frac{5}{16}$ in diameter studs. Cemjo steel faced joint washers are fitted. The radius through which the axis of the port is turned from the valve seat insert is 0·90 in. The inlet ports are siamesed, and the radius through which the axes are turned is about 1·94 in. Although this is a fairly small radius for a port whose section is $1\frac{5}{16}$ in square, the angle turned through is only 70 deg and the maximum speed is only 2,800 r.p.m., so there is little doubt that the breathing capacity is more than adequate. Plain Cemjo washers are interposed between the joint faces on the head and the D.T.D.424 cast aluminium inlet manifold. The two castings are bolted together under the inlet riser pipe to form a hot spot.

Considerable care has been taken with the engine breathing system, not only to ensure thorough ventilation of the crankcase, but also to keep out foreign matter. This is particularly important for many overseas markets. A small oil bath air filter with a gauze



Seventeen bolts hold down the cylinder head, and seven more effectively secure the manifolds



A hot spot is formed where the inlet and exhaust manifolds are bolted together

element is mounted on top of the oil filler cap which is on the rocker cover. This ensures that air entering the rocker chamber and crankcase is uncontaminated. The air outlet pipe is taken from the tappet chamber cover to the air intake above the carburettor.

A Zenith 36 VIS downdraught carburettor is fitted on an asbestos and steel insulating block on the manifold. The choke diameter is 32 mm and the jet sizes and other dimensions are: main, 90; slow running, 55; compensating jet, 105; needle seat angle, 40 deg. Fuel is delivered to the carburettor by an A.C. diaphragm-type pump. It incorporates a hand priming lever and is actuated by a camshaft eccentric between numbers 3 and 4 cylinders. A removable glass sediment bowl is fitted to the pump and the fuel is drawn from a 14 gallon tank. An A.C. oil bath air filter is connected by a length of rubber hose to the carburettor air intake.

Water pump and cooling system

A canvas reinforced rubber Vee-belt supplied by Goodyear Tyre and Rubber Company (Gt. Britain) Ltd. or Ferodo Ltd. transmits the drive to the water pump. The driving pulley pitch diameter is 7·035 in and the diameter of the driven pulley is such that the pump speed is 1·35 times engine speed. A Vee-angle of 40 deg has been adopted for the belt, the cross sectional dimensions of which are $\frac{3}{4}$ in wide by $\frac{5}{8}$ in thick.

A conventional water pump arrangement has been used, in fact, it is the same as that of the Fordson Major engine. The rotor diameter is $2\frac{7}{8}$ in. It is pressed on to the spindle, with its shroud to the rear, so that as the pump body is spigoted into the front of the cylinder block, there is no need for a rear wall to close the end of the pump. Between the tubular type, two-row ball bearing and the rotor is a water seal and thrower. Axial location is effected by a wire clip pushed through a slot in the top of the nose piece of the body into a circular section groove, half of which is machined in the bearing outer race and the other half of which is in the housing.

At the front end of the spindle the pulley boss is pressed on. A flanged dished pressing is used, as before, the flange forming one side of the rim channel, and the other side of the channel is pressed on and spot welded. The pulley assembly is spigoted and bolted on the boss. A four-bladed, 18 in diameter fan is used in the countries where temperatures exceeding 90 deg F are likely to be experienced, and two-bladed units are supplied for use elsewhere. The four-blade fan is made from two pieces, each forming two blades, welded one on top of the other, with their axes at right angles. A rib is pressed into the centre of each piece to furnish the necessary stiffness in the longitudinal direction.

Cooling is effected by a fin and tube radiator having a frontal area of

Oil from the relief valve at the front end of the gallery is discharged on to the half speed wheel

$439\frac{1}{2}$ in^2. Its block dimensions are 3 in thick by 19·75 in wide by 22·5 in high. The system is not pressurized. From the radiator, water passes to the pump and into the cylinder block. From there, it goes through ducts under the sparking plug bosses and exhaust ports into the cylinder head. A thermostat valve is carried in an extension of the head casting and the water outlet pipe is bolted on above it. The thermostat valve begins to open at 156 to 165 deg F and should be fully open at 180 deg F.

Oil pump and lubrication system

The gear type oil pump, running at half engine speed, is the same as the one used in the Fordson Major engine. It is housed in the same way in a blister on the right-hand side of the crankcase casting. Its vertical drive spindle is driven by a spiral gear carried on another spindle, the axis of which is parallel to that of the crankshaft. Each end of this horizontal spindle is carried in a ball bearing housed in the crankcase blister. This spindle is driven in the same way as the injector pump drive spindle in the diesel tractor engine by a helical gear that meshes with the one behind the half speed wheel. This gear is of cast iron and is keyed on the front end of the spindle and secured by a split pinned slotted nut.

An aluminium sump with two integrally cast transverse baffles is employed. Its capacity is 12 pints. Oil is drawn from the well in the sump, through a strainer, to the pump. The inlet port in the pump is arranged to form an oil trap so that the unit remains primed even when the engine is stationary. Lubricant is fed from the pump through a short duct to a full-flow filter mounted on the side of the crankcase. The filtered lubricant then passes through a duct in the crankcase web that carries number 2 main journal to the main gallery on the opposite side of the crankcase. A pressure relief valve is screwed into the front end of the gallery and the blow-off oil lubricates the timing gears. A small bleed hole ensures that even when the valve is closed there is always a supply of oil to the gears.

Each crankshaft main journal is served by ducts from the gallery, and drillings in the crankshaft carry the lubricant from the main journals to the big ends. The small end is splash lubricated. From each crankshaft main journal, a passage is drilled to the adjacent camshaft bearing. The centre journal of the camshaft is drilled chordwise in such a way that as it rotates oil is intermittently fed up a vertical drilling through the cylinder block and head to the centre rocker shaft pedestal. A rubber ring forms a seal round the vertical drilling at the junction between the cylinder block and head. From the pedestal the lubricant is taken into the hollow rocker shaft in which there are radially drilled holes to distribute it to the rocker bearings. Some of the oil passes out through holes in the top of the rockers to run down to the push rod and valve ends.

Ignition and electrical equipment

Lucas ignition and electrical equipment is used throughout. A Lucas DM4 distributor, with a vacuum and centrifugal automatic advance and retard mechanism, is driven from the top of the vertical spindle on the lower end of which is the oil pump. It is served by a B 120-LO-ER coil. The sparking plugs supplied are Champion 14 mm type with the gap set at 0·028-0·033 in. The dynamo is a C45-PV5-LO 12 volt, shunt wound unit capable of delivering a maximum of 22 amps. It is driven at 1·35 times engine speed. An RB106-1 voltage regulator is employed and the battery is a 12 volt unit of 57 amp-hr capacity at a 20 hour rate manufactured by Lucas to a Ford specification. An M45G-L7 starter with an inboard drive is fitted and the pinion has 10 teeth.

MECHANIZED MANUFACTURE

Important American Production Developments

ONE of the most outstanding production developments of recent years in the American automobile industry is that for which the term "automation" has been coined. Briefly, this may be defined as the alignment of a number of machines in a production sequence with conveying equipment, feeders and positioning devices to make a line of dissimilar operations completely automatic and continuous. There are two main reasons for this development. The first is to reduce the labour force necessary to produce a given output; this is a much more potent reason for development in the U.S.A. than in this country. Secondly, "automation" has been developed to allow fuller use to be made of the production potentials of modern machine tools.

The automatic in-line transfer machines now used by the automobile industry in this country are a stage in the trend towards continuous automatic manufacturing equipment, but American developments have been carried much further, particularly as regards machine loading and inter-machine handling of components. For example, there is an installation at the Cleveland, Ohio, plant of the Ford Motor Company so mechanized that cylinder block and cylinder head castings are scarcely touched by hand during the machining sequence.

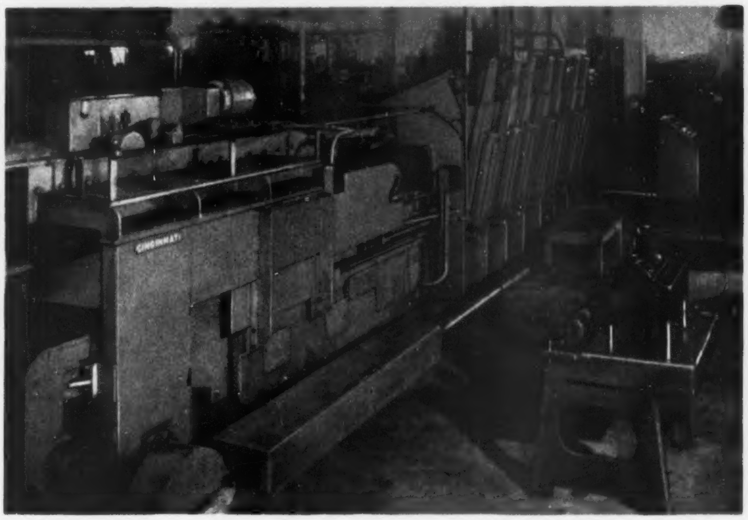


Fig. 1. Automatic broaching machine for Ford six-cylinder blocks

Cylinder block machining

In machining sequence that includes so many operations as are necessary on a cylinder block, it is inevitable that there will be an appreciable difference in the cycle times for the fastest and slowest operations. Therefore, work must be allowed to build up from the faster operations or the machines for the slower operations must be duplicated. In the Ford plant the latter alternative has been adopted. This decision in some respects created additional difficulties, which have been successfully overcome.

The first machining operation on the cylinder block is carried out on a huge horizontal broaching machine, see Fig. 1, which machines the head and sump faces, the main journals and the main bearing locks at one pass. This machine has an output capacity of 197 blocks per hour. The next operation has a considerably longer cycle time, so to maintain a balanced production the machines for this operation are duplicated. Essentially the system is such that each block as it leaves the broaching machine is automatically transferred to whichever of the succeeding machines is waiting for work.

Fig. 2 shows the discharge end of the broaching machine. The casting is either picked up by the transfer bar at the right of the illustration or is shuttled broadside to meet the conveyor for the alternative machine. Both conveyors and the shuttle conveyor are shown in Fig. 3.

Another arrangement for feeding

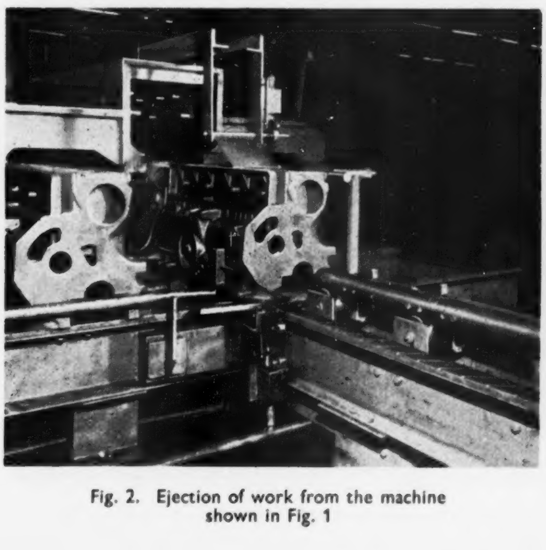


Fig. 2. Ejection of work from the machine shown in Fig. 1

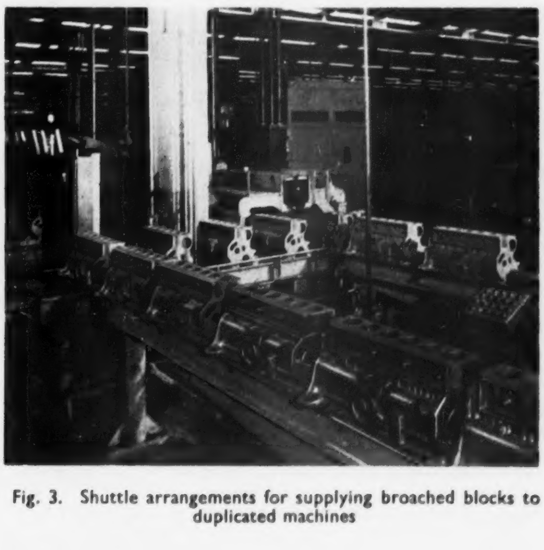


Fig. 3. Shuttle arrangements for supplying broached blocks to duplicated machines

Fig. 4. Arrangement for automatically turning and feeding blocks to duplicate machines

Fig. 5. Turntable station for rotating blocks through 90 deg

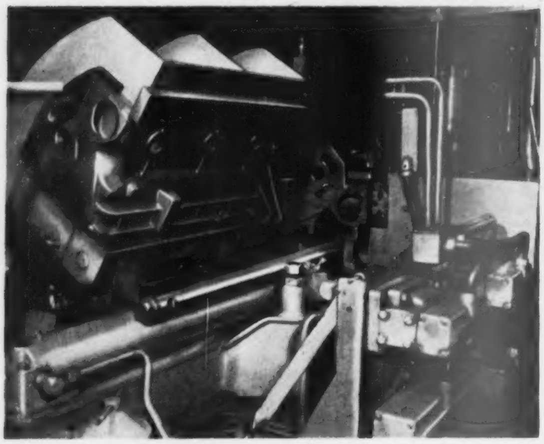


Fig. 6. Loading device which automatically tilts through 90 deg to bring the component into the working position

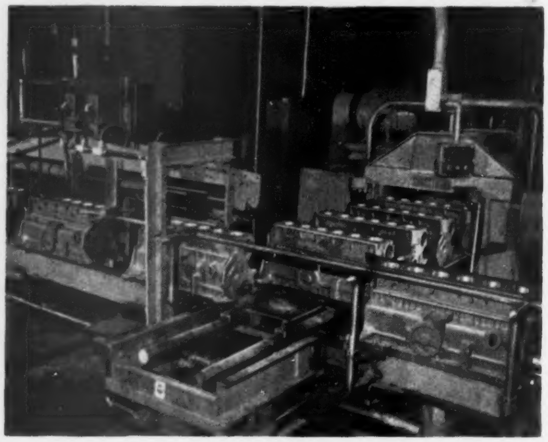


Fig. 7. Overhead transfer bar and broadside shuttle for feeding a milling machine

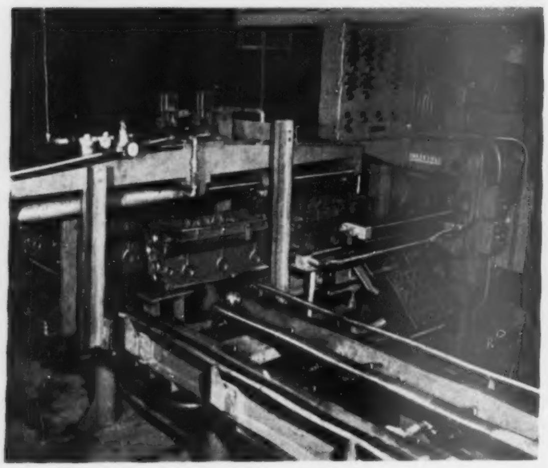


Fig. 8. Another method of feeding blocks to a milling machine

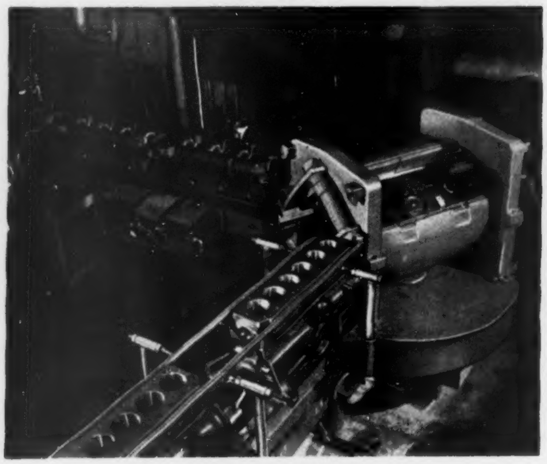


Fig. 9. Automatic two-directional turn-over device

cylinder blocks to duplicate machines is shown in Fig. 4. As can be seen from the illustration, the blocks are fed from the machine at the right towards the machines at the left. Directly in line with the first of the duplicated machines is a turntable of the type shown in Fig. 5 which automatically turns the casting through 90 deg. Then an electronic controller selects whether the work is fed to the first machine or to the second.

An interesting automatic loading device is shown in Fig. 6. The block enters the loading device with the head face uppermost but it must be turned through 90 deg for the operation that is to be performed. When the block is in position, the loading device is automatically tilted to bring the casting to the working position. On the completion of the operation, the block is automatically returned to its original position with the head face uppermost and is then ejected from the machine.

Another arrangement for feeding duplicate machines is shown in Fig. 7. Direction of work travel is from left to right. An overhead shuttle bar at the left moves the work to a position from which it can either be moved at right angles to the first of two Sundstrand milling machines or, if this machine is fully supplied, it will move forward to a second Sundstrand machine.

The automatic feeding arrangements for an Ingersoll machine are shown in Fig. 8. At this point two cylinder blocks are pushed forward simultaneously towards the loading station of the machine. This pushes the leading block into a position where it can be picked up by the transfer bar of the machine tool, while the second block is moved forward to make room for the block that is following.

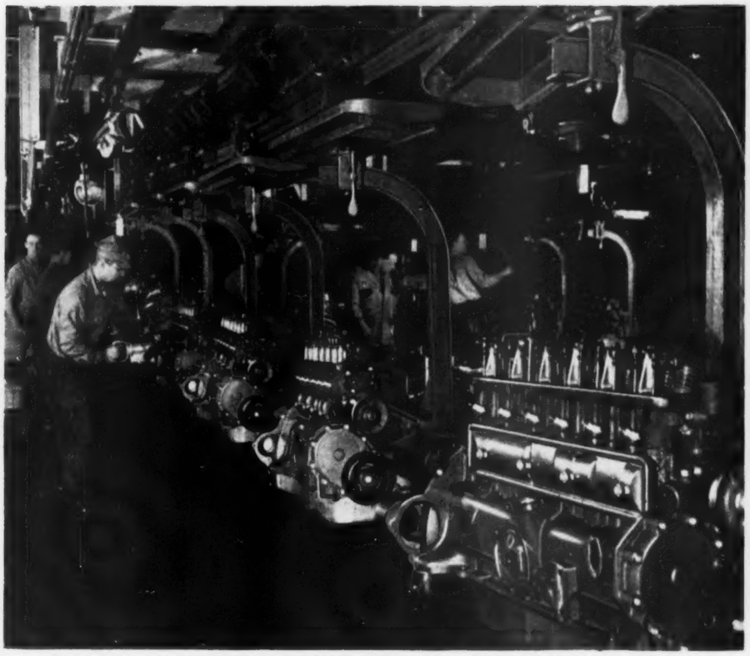


Fig. 10. The engine assembly conveyor

An interesting two-directional automatic device is shown in Fig. 9. It is installed in the cylinder block line between the end of the machining sequence and the point at which the block is transferred to the assembly conveyor. The block leaves the final machine in the sequence with the head face uppermost and the front end leading, but for the early assembly operations it is necessary to have the sump face uppermost and the rear end leading. This change of position is effected automatically in the device shown in Fig. 9. The necessary compound indexing motion is effected quickly and without shock. When the turnover is completed, another automatic shuttle extracts the block from the device and delivers it to the assembly conveyor.

In contrast to practice in this country and also to conventional American practice, an overhead conveyor is used for engine assembly, see Fig. 10. At the start of the assembly operations a carrier arm, mounted on an overhead conveyor, is attached to the cylinder block. It is not removed until the engine has been tested and is ready for shipping. Engines on test are shown in Fig. 11.

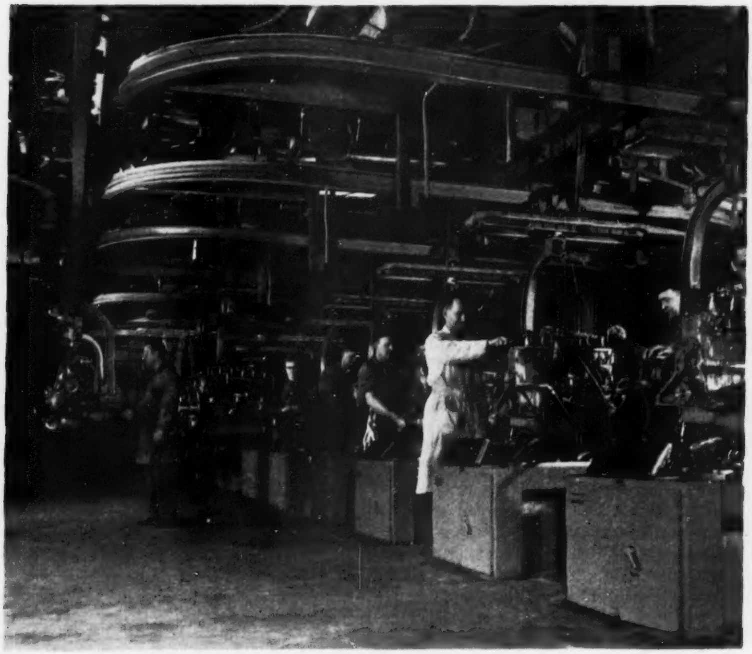


Fig. 11. The engine test section

Coil spring production

Ford Motor Company also employ continuous automatic manufacturing methods for the production of coil springs. Basically, the line comprises a rod furnace for heating the rods before coiling, a group of roll formers for flattening the rod ends to give the square ends of the springs, hot roll coilers, quench tanks and wheelabrators for cleaning the springs after heat treatment.

Transfer of the rods from the furnace to the roll formers is effected automatically by means of a long air cylinder that pushes the rods out of the furnace into the roll former. As the rod leaves the roll former it is picked up by an arm that feeds it to the roll coiler. As soon as the coiling is completed the spring is ejected from the coiler to be picked up by another arm and fed to the quench system.

An important feature of this line is

an indexing mechanism that turns every spring over to ensure that the open end of the spring is uppermost during the quenching operation. This is necessary because of the danger that the spring will be crushed unless the open end is in the upright position.

It must be emphasized that the spring is taken direct from the coiling rolls to the quench. This means that with only one heating and after all the operations have been performed, the spring temperature is still high enough to ensure correct metallurgical treatment without any re-heating. In a conventional line for coil spring production it would be necessary to install expensive furnaces for re-heating the springs after the coiling operation. When the springs leave the quench they pass to a paddle feeder which turns them into position and then feeds them in proper sequence to one of three wheelabrators. The daily production rate of this line is greater by 4,000 sets of springs than that of any comparable line, and at the same time the man power for the line has been reduced from 248 to 120.

A pioneer of automation, C. F. Hautau, of Hautau Engineering Company, Detroit, Michigan, has pointed out that most modern machine tools are provided with satisfactory feeding and loading devices for manual operation, but they seldom have devices that will properly index, orient and position the work as it comes from the preceding machine. In effect, automation is the science of developing such devices. It is claimed that systems already in use in America have shown savings in the order of 50 per cent in production costs.

There are other benefits which are of a nature that cannot be included in actual cost figures. For example, when a line is completely automatic it is possible to reduce the tolerance spread for mating parts, since such a line must have inspection devices built into it at the processing stations. This possibility of eliminating the human element in inspection is important.

FATIGUE TESTS

The Statistical Interpretation of Results

A PAPER entitled "On the Statistical Interpretation of Fatigue Tests," by A. M. Freudenthal and E. J. Gumbel has been published in the *Royal Society Proceedings, Series A,* February 10, 1953. Results of fatigue tests performed under assumedly identical conditions show considerable scatter. Thus, if n specimens are subjected to a sequence of stress cycles of the same amplitude S, they break at varying numbers of cycles; these numbers N taken in decreasing order, and the frequencies of survival at each number, determine, for each stress level S, a characteristic cumulative frequency distribution $1(N)^S$, the *survivorship function.*

By formulating as a problem of extreme values the phenomenon of consecutive fatigue fractures of the weakest within a finite (large) set of specimens, the statistical theory of extreme values can be applied to the interpretation of the observed frequencies of survival at any stress amplitude. If, in first approximation, it is assumed that the probability of survival reaches unity only for $N=0$ (no "sensitivity threshold" in N), the survivorship functions are reproduced by the "third asymptotic probability function of smallest values," which is represented on extremal probability paper by a straight line relation between a reduced statistical variate y and $\log_{10}N$. Methods are presented for the computation of the two parameters of the survivorship function $1(N)^S$ from a set of fatigue data. The fit between the computed theoretical straight lines and the test results is satisfactory for fatigue tests of copper, aluminium and a high-strength structural aluminium alloy. Reasons are suggested for cases where agreement between the proposed theory and observations is not satisfactory.

Results of fatigue tests can be used as statistical observations if the following prerequisites are fulfilled: (1) A constant number of specimens, at least twenty, should be tested for each stress level, at as many stress levels as possible. The same machine should be used throughout. (2) The stress levels should be kept constant, sufficiently separated to make the differences significant, and sufficiently near to allow for the construction of the survivorship function for constant numbers of cycles N and variable stress levels S. (3) No series of test results are admissible which lead to intersections of neighbouring functions. (*M.I.R.A. Abstract* No. 6305.)

FATIGUE

IN an article entitled "Fatigue of Ferrous Metals — A Phenomenological Approach," by H. L. Cox, in *Iron and Steel,* February 1953, an attempt is made to marshal the experimental evidence and to present the subject in a phenomenological manner and to explain how a fatigue crack starts or propagates. The conception of fatigue embodied in the conventional S-N diagram is explained as also is the necessity to establish that, for a given material, the same value of fatigue limit must be found for all types of specimen or methods of test.

The present state of knowledge regarding fatigue of steels may be briefly summarized as follows: The limiting fatigue strength of any steel under alternating uniform uniaxial stress is an intrinsic property of the material, and its magnitude (total range) is usually slightly greater than the tensile strength. The fatigue strength under alternating shear stresses q_0 is about 60 per cent of the fatigue strength under alternating direct stresses f_0, and under combined stresses f and q the fatigue strength is given by the relation $(f/f_0)^2+(q/q_0)^2=1$. As this relationship suggests, the true criterion of fatigue strength almost certainly concerns not the stress value at a single point, but some average value of stress throughout some small volume, possibly associated with grain size.

In accordance with this theory, the fatigue strength must be influenced by the gradient of stress in the region of maximum stress. If results of tests are represented in terms of range of nominal maximum stress and stress gradient computed by elastic theory, the effect of the gradient may be exaggerated because the actual range of maximum stress may be reduced by continuing plastic deformation (hysteresis), but this will also reduce the stress gradient.

Mean loading of a moderate value has relatively little effect, and practically none if it differs in type from that of the range of loading. Heavy mean compressive stress, however, may improve the fatigue strength of notched specimens by preventing propagation of fatigue cracks. Fatigue cracks usually propagate from regions of maximum stress following fairly well-defined paths, but in some cases may stop and final failure may result elsewhere. A statistical hypothesis for the phenomenon, based on this uncertainty of failure is put forward and discussed. (*M.I.R.A. Abstract No.* 6304.)

ACCELERATION

Its Influence on Fuel Consumption and Journey Time

Dr. J. G. Lavender and Dr. C. R. Webb

THIS paper describes an analytical method, based on test results and empirical relationships, for determining to what extent good acceleration in a vehicle reduces journey time, and what effect it has upon fuel consumption. Arkus-Duntov (Ref. 1) has investigated vehicle resistance in respect of a racing car, from the point of view of finding the relation between engine power and maximum speed. Webb (Ref. 2) dealt with the effect of road gradient on journey time and fuel consumption, but only steady road speeds were considered. This investigation studies the same vehicle considered in Ref. 2, from the point of view of acceleration and deceleration on level roads.

The effects of adjustments in ignition setting, mixture strength and compression ratio on fuel consumption have been dealt with in a separate paper (Ref. 3); the present work is based on engine test characteristics obtained with the manufacturers' settings. The suggested procedure is applied in this paper to an economical family saloon car, but is of general application to all road vehicles. It is shown that rapid acceleration is wasteful of fuel and only time saving to a limited extent. The advantages of fitting both an overdrive and a freewheel are also shown.

Vehicle data

Information concerning the particular vehicle under consideration, so far as required in the calculation, is given below:

Engine capacity 68·7 cu in
Compression ratio 6·5
Unladen weight 18·6 cwt
Laden weight (driver and 3 passengers) 23 cwt
Projected frontal area 19·4 sq ft
Forward gear ratios:
- 1st 21·60
- 2nd 13·06
- 3rd 8·223
- 4th 5·375 (direct-drive)
- *5th 4·300 (overdrive).

*For comparative purposes a 5th gear overdrive has been assumed, giving a ratio 25 per cent above the normal 4th gear direct-drive.

Engine speed in 4th gear at 50 m.p.h. 3,450 r.p.m.
Moment of inertia of engine moving parts 7 lb-ft²
Moment of inertia of road wheels, back axle, differential and transmission 170 lb-ft²
Tyre pressure 24 lb/in².

Engine power

Fig. 1 indicates test curves of engine torque (b.m.e.p.) against engine speed (r.p.m.) at various throttle openings (percentage of maximum opening). Zone 1 represents the conditions when the engine is driving and Zone 2 when the engine is being driven on the overrun. The complete figure is bounded by the speed lines of 500 r.p.m., representing the idling speed of the engine and the minimum driving speed, and 4,200 r.p.m. representing the maximum allowable engine speed. The range of road speeds considered in the various gears has been limited by these engine speeds.

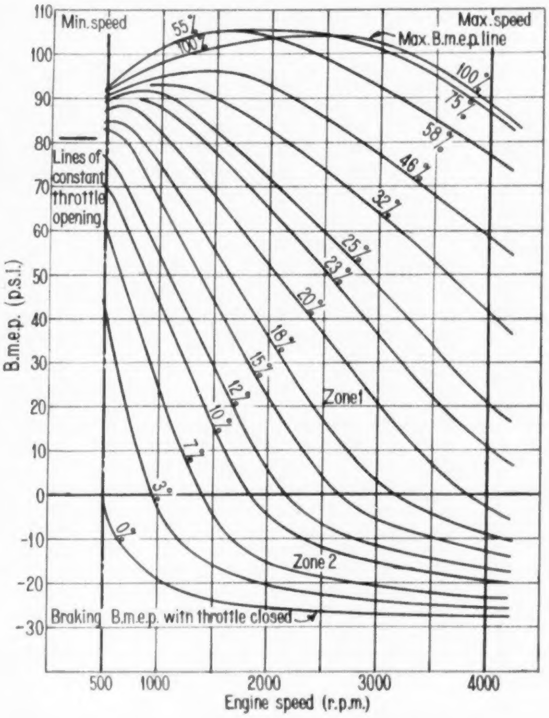


Fig. 1. Engine torque at various throttle openings

It should be noted that the maximum driving torque line does not always correspond with 100 per cent throttle opening. In fact, the maximum torque is given by 55 per cent throttle opening between 500 and 1,600 r.p.m., 75 per cent opening between 1,600 and 2,500 r.p.m. and 100 per cent opening from 2,500 to 4,200 r.p.m. The absolute maximum torque occurs at 2,000 r.p.m. with 75 per cent throttle opening.

In Zone 2, on the overrun, the negative b.m.e.p. represents the effective braking torque that can be supplied by the engine at different speeds and throttle openings. The maximum braking torque is always obtained with the throttle closed (0 per cent) and the value of this torque is almost constant between speeds of 2,500 and 4,200 r.p.m.

Fuel consumption

The fuel consumption (lb/hr) over the engine speed range of 500 to 4,200 r.p.m., at various throttle openings, is shown in Fig. 2. Zone 1, above the line of zero output, represents the driving conditions and Zone 2, below this line, represents the overrun conditions. In nearly all cases with fixed throttle openings, the fuel consumption on the overrun shows little increase with speed.

Tractive resistance of vehicle

When the vehicle is accelerated on a level road in still air conditions the total resistance to be overcome by the engine may be considered as the sum of four resistance components. These are (i) the rolling resistance, (ii) the air resistance, (iii) the frictional resistance of the transmission, and (iv) the inertia resistance of the vehicle and rotating parts.

The components (i) and (ii) are calculated from accepted empirical equations for any particular road speed.

(i) *Rolling Resistance*

Reference 4 gives the following equation which is applicable to small passenger cars:

$$R_r = W \frac{245}{p^{0.64}} + \frac{V^{3.7}}{396p^{2.08}} \quad \ldots\ldots (1)$$

where R_r = rolling resistance in lb,
W = vehicle weight in tons,
V = speed in m.p.h.,
p = tyre pressure in lb/in².

(ii) *Air Resistance*

The air resistance of cars of modern shape (Ref. 4) is given by:

$$R_a = 0{\cdot}0017\, AV^2 \quad \ldots\ldots (2)$$

where R_a = air resistance in lb,
A = projected frontal area in ft².

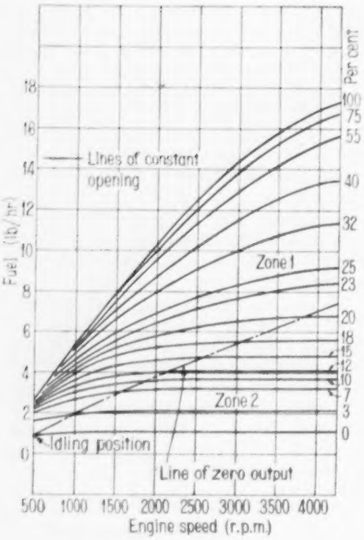


Fig. 2. Fuel consumption at various throttle openings

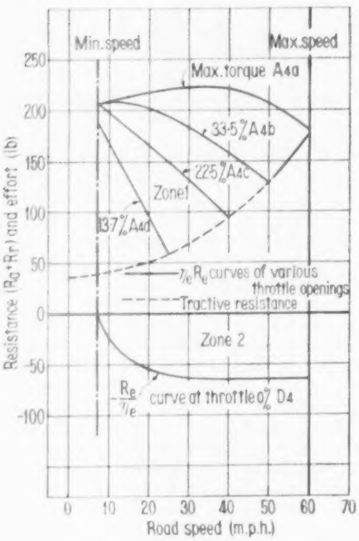


Fig. 3. Tractive effort and resistance in 4th gear

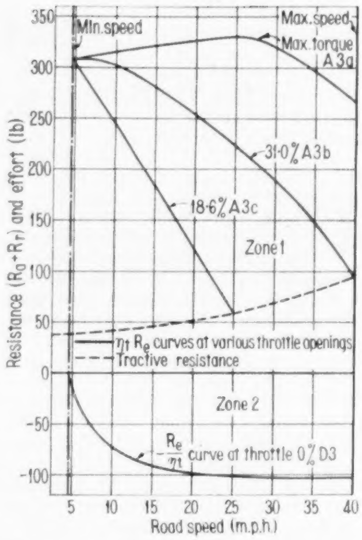


Fig. 4. Tractive effort and resistance in 3rd gear

(iii) *Frictional Resistance of Transmission*

A transmission efficiency factor η_t, is used to account for the power loss, in either direction, through the clutch, gear box, propeller shaft and rear axle gearing. In 4th gear (direct drive) and in overdrive, η_t is taken as 95 per cent and in other gears as 92 per cent.

(iv) *Inertia Resistance*

The total inertia effect of the vehicle, including road wheels, transmission, gearing and the engine, is expressed as the inertia of an equivalent weight, E, having the same linear motion as the vehicle. The inertia resistance is then given by:

$$R_f = \frac{E}{g} f \quad \text{.....................(3)}$$

where R_f = inertia resistance in lb,
f = linear acceleration in ft/sec^2.

Alternatively,

$$f = k R_f \quad \text{.....................(4)}$$

where $k = \frac{g}{E}$.

The value of the equivalent weight of the vehicle varies according to the gear in use and, from the vehicle data given, the values of E and k have been calculated as follows:

5th gear (overdrive)
$E_5 = 2847{\cdot}2$ lb, $k_5 = 0{\cdot}01129$
4th gear
$E_4 = 2888{\cdot}6$ lb, $k_4 = 0{\cdot}01113$
3rd gear
$E_3 = 3117{\cdot}2$ lb, $k_3 = 0{\cdot}01032$
2nd gear
$E_2 = 3722{\cdot}2$ lb, $k_2 = 0{\cdot}00864$
Freewheeling
$E_0 = 2718{\cdot}2$ lb, $k_0 = 0{\cdot}01183$.

The case of freewheeling has been calculated assuming the engine disengaged from the gear box.

Arkus-Duntov (Ref. 1) uses an equivalent weight of the chassis rotating parts of 160 lb, compared with the figure of 140 lb used here. He neglects altogether the equivalent weight of the engine rotating parts, which in the present case is 170 lb in 4th gear and 1,004 lb in 2nd gear.

Tractive effort

When the vehicle is being driven by the engine the four components of tractive resistance must be balanced by the tractive effort produced by the engine. Thus,

$$(R_r + R_a + R_f) = \eta_t R_e \quad \text{..........(5)}$$

where R_e (lb) is the tractive effort produced by the engine and $\eta_t R_e$ is the available tractive effort at the rear wheels.

Fig. 3 indicates the available tractive effort ($\eta_t R_e$) in 4th gear at particular throttle openings over the range of road speeds attainable in this gear. The tractive resistance line represents the tractive effort required to propel the vehicle at constant speed and is obtained from equations (1) and (2) using the vehicle data quoted.

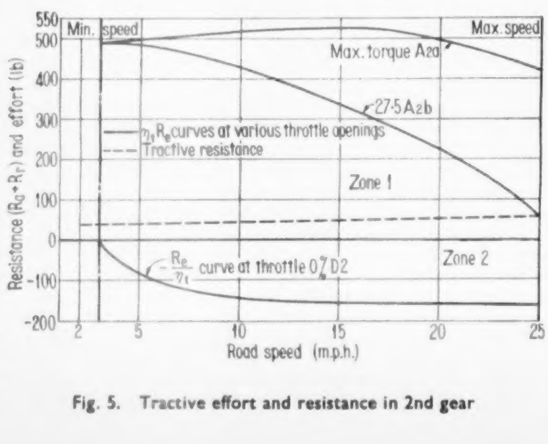


Fig. 5. Tractive effort and resistance in 2nd gear

The points of intersection between the constant throttle lines and the tractive resistance line indicate the steady speeds that may be attained with these fixed throttle settings. For instance, in 4th gear, with a constant throttle opening of 22·5 per cent, the vehicle travels at a steady speed of 40 m.p.h.

The line forming the lower limit of Zone 2 shows the variation in engine braking resistance with road speed and is obtained from the closed throttle line on Fig. 1. It will be noted that as the drive is in the reverse direction on the overrun, this line is plotted as R_e/η_t. In this case, equation (5) does not represent the motion of the vehicle.

Figs. 4, 5 and 6 indicate similar curves for 3rd, 2nd and 5th gears respectively, and in all cases throttle openings have been selected, where possible, to produce steady road speeds of 25, 40, 50 or 60 m.p.h. The braking resistance, used later in the calculations, is with the engine in gear and the throttle closed, apart from the case of freewheeling.

The following key indicates the nomenclature referring to the effort curves plotted in Figs. 3 to 6, and all the cases, except those shown thus*, are used in later figures.

- A4a, 4th gear, engine driving, maximum torque.
- A4b, 4th gear, engine driving, 33·5 per cent throttle opening.
- A4c, 4th gear, engine driving, 22·5 per cent throttle opening.
- A4d, 4th gear, engine driving, 13·7 per cent throttle opening.
- D4, 4th gear, overrun, throttle closed.
- A3a, 3rd gear, engine driving, maximum torque.

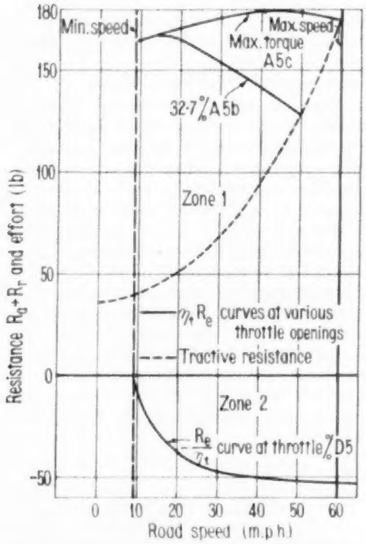


Fig. 6. Tractive effort and resistance in overdrive

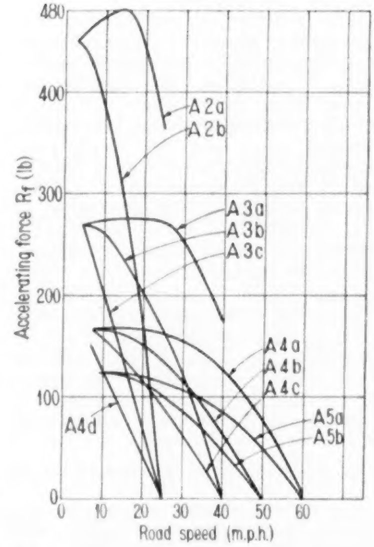


Fig. 7. Accelerating force in various cases

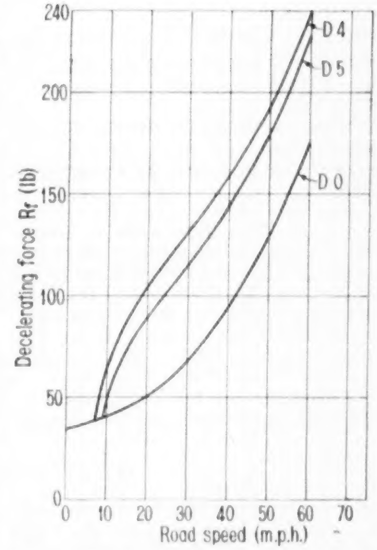


Fig. 8. Decelerating force in various cases

A3b, 3rd gear, engine driving, 31·0 per cent throttle opening.

A3c, 3rd gear, engine driving, 18·6 per cent throttle opening.

*D3, 3rd gear, overrun, throttle closed.

A2a, 2nd gear, engine driving, maximum torque.

A2b, 2nd gear, engine driving, 27·5 per cent throttle opening.

*D2, 2nd gear, overrun, throttle closed.

A5a, 5th gear, engine driving, maximum torque.

A5b, 5th gear, engine driving 32·7 per cent throttle opening.

D5, 5th gear, overrun, throttle closed.

D0, Freewheeling, engine idling, throttle closed.

In all gears, apart from 5th, the maximum torque throttle settings produce a greater tractive effort than that required to propel the vehicle at the steady road speed limited by the maximum engine speed of 4,200 r.p.m.

Accelerating force

In Figs. 3 to 6 it will be seen that at a particular road speed and throttle opening, the vertical distance between the tractive effort line and the tractive resistance line will give the excess force available for acceleration. Equation (5) may be rewritten in the form

$$R_f = \eta_t R_e - (R_r + R_a) \qquad \ldots\ldots\ldots\ldots(6)$$

For example, from Fig. 3, in 4th gear at 40 m.p.h. with maximum torque, there is available an accelerating force of 126 lb, and with 33·5 per cent throttle opening there is an excess force of 64 lb. With 22·5 per cent throttle opening, there is no excess force available, and the vehicle will travel at the constant speed of 40 m.p.h.

Fig. 7 shows the variation in accelerating force with road speed in the different gears and with different throttle settings. In each gear with the maximum torque settings (A2a, A3a, etc.) the accelerating force initially tends to increase with increase in road speed, whereas with the smaller throttle openings (A2b, A3c, etc.) the accelerating force continually decreases.

On the overrun the decelerating force is estimated as the sum of the engine braking resistance and the tractive resistance and may be expressed in the form

$$R_f = \frac{R_e}{\eta_t} - (R_r + R_a) \qquad \ldots\ldots\ldots\ldots(7)$$

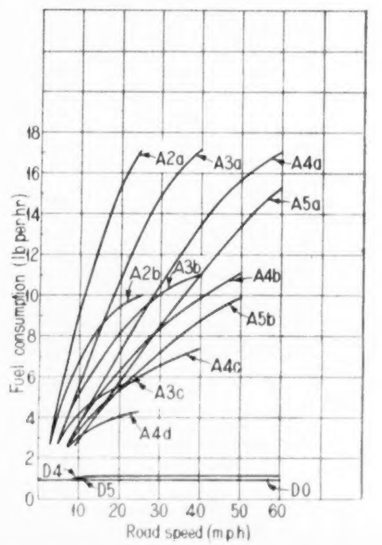


Fig. 9. Fuel consumption in various cases

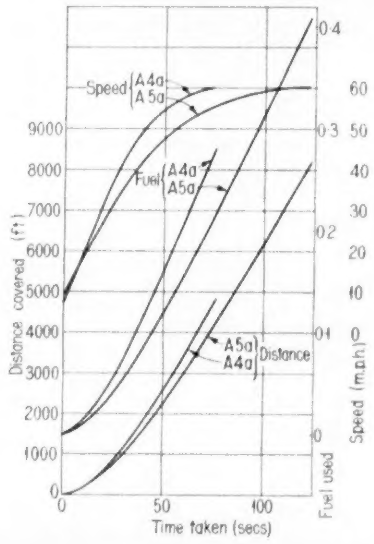


Fig. 10. Acceleration characteristics to 60 m.p.h.

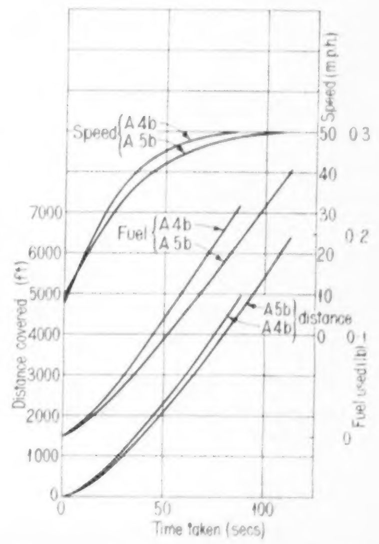


Fig. 11. Acceleration characteristics to 50 m.p.h.

In this equation $\frac{R_e}{\eta_t}$ is negative, $(R_r + R_a)$ is positive as in equation (6) and R_f becomes negative to represent deceleration.

When the vehicle is allowed to freewheel there will, of course, be no engine braking so that the equation (7) becomes

$$R_f = -(R_r + R_a) \quad \dots\dots\dots\dots\dots(8)$$

Fig. 8 shows three cases in which the vehicle is decelerated from 60 m.p.h. (D5, D4 and D0). The curve D0 is continued until the vehicle is at rest and at this point the decelerating force is obtained from equation (1) as:

$$R_f = -R_r = -\frac{245W}{p^{0.64}} \quad \dots\dots\dots(9)$$

Curves D4 and D5 are continued to the road speeds equivalent to an engine speed of 500 r.p.m. and intersect the curve D0 at these points. At the higher road speeds these curves follow similar shapes to that of the freewheeling curve, being raised above the latter by an amount equivalent to the engine braking effect. At the lower road speeds, however, as the engine braking effect rapidly decreases (Fig. 1), the curves D4 and D5 drop sharply on to the freewheeling curve, meeting it where the engine braking torque is zero (i.e. 500 r.p.m.).

The fuel consumption during periods of acceleration and deceleration is obtained from Fig. 2 and is shown in Fig. 9 plotted on a basis of road speed. The deceleration curves D4 and D5 lie extremely close together whereas the curve D0 is a horizontal straight line representing the fuel consumption of the engine when idling.

Vehicle acceleration

The first three columns of Table I show, for 4th gear and maximum torque conditions, the values of tractive effort and tractive resistance at road speeds between 7·2 and 60 m.p.h., column 4 indicating the equivalent engine speeds. Column 5 gives the values of R_f, the accelerating force, calculated from equation (6). The fuel consumption at each particular road speed is given in column 6. The change in speed (ft/sec) is tabulated in column 7 and the average accelerating force available during these speed changes is given in column 8 (obtained from Fig. 7). Using equation (4), with the

TABLE I. 4th GEAR ACCELERATION—MAXIMUM TORQUE k4 = 0·01113

$\eta_t R_e$ (lb)	R_a+R_r (lb)	Road Speed (mph)	Engine Speed (rpm)	R_f (lb)	Fuel Rate (lb/hr)	Speed Change (ft/sec)	Mean R_f (lb)	Mean f (ft/sec²)	Time Taken (secs)	Aggregate Time Taken (secs)	Mean Fuel Rate (lb/hr)	Fuel Used (lb)	Aggregate Fuel Used (lb)	Average Speed (mph)	Distance Covered (ft)	Aggregate Distance Covered (ft)
206	39	7·2	500	167	2·6					0			0			0
						4·11	167	1·860	2·21		3·15	0·0019		8·6	27·9	
208	41	10	690	167	3·65					2·21			0·0019			27·9
						7·33	168	1·870	3·91		4·55	0·0050		12·5	71·6	
213	45	15	1035	168	5·5					6·12			0·0069			99·5
						7·33	167	1·860	3·92		6·30	0·0068		17·5	100·8	
217	51	20	1380	166	7·05					10·04			0·0137			200·3
						7·33	164	1·827	3·99		7·95	0·0089		22·5	132·0	
220	58	25	1725	162	8·8					14·03			0·0226			332·3
						7·33	158·5	1·768	4·13		9·45	0·0108		27·5	166·6	
222	68	30	2070	154	10·35					18·16			0·0334			498·9
						7·33	149	1·660	4·39		11·15	0·0136		32·5	209·5	
223	80	35	2410	143	11·75					22·55			0·0470			708·4
						7·33	135·5	1·510	4·83		12·65	0·0170		37·5	265·5	
220	94	40	2760	126	13·5					27·38			0·0640			973·9
						7·33	116	1·292	5·64		14·10	0·0221		42·5	351·0	
215	110	45	3100	105	14·7					33·02			0·0861			1324·9
						7·33	92	1·024	7·12		15·20	0·0301		47·5	496·0	
206	128	50	3450	78	15·7					40·14			0·1162			1820·9
						7·33	62	0·691	10·55		16·12	0·0473		52·5	813·0	
195	150	55	3790	45	16·45					50·69			0·1635			2633·9
						7·33	26	0·289	25·20		16·80	0·1178		57·5	2125·0	
180	175	60	4140	5	17·1					75·89			0·2813			4758·9

TABLE II. 4th GEAR DECELERATION—THROTTLE CLOSED k4 = 0·01113

Overrun R_e/η_t (lb)	R_a+R_r (lb)	Road Speed (mph)	Engine Speed (rpm)	R_f (lb)	Fuel Rate (lb/hr)	Speed Change (ft/sec)	Mean R_f (lb)	Mean f (ft/sec²)	Time Taken (secs)	Aggregate Time Taken (secs)	Mean Fuel Rate (lb/hr)	Fuel Used (lb)	Aggregate Fuel Used (lb)	Average Speed (mph)	Distance Covered (ft)	Aggregate Distance Covered (ft)
—65	175	60	4140	—240	1·10					0			0			0
						7·33	227	2·530	2·90		1·10	0·00081		57·5	245·0	
—65	150	55	3790	—215	1·10					2·90			0·00081			245·0
						7·33	204	2·270	3·23		1·10	0·00098		52·5	249·0	
—65	128	50	3450	—193	1·10					6·13			0·00179			494·0
						7·33	184	2·050	3·57		1·10	0·00109		47·5	248·5	
—65	110	45	3100	—175	1·10					9·70			0·00288			742·5
						7·33	167	1·860	3·94		1·10	0·00120		42·5	246·0	
—65	94	40	2760	—159	1·10					13·64			0·00408			988·5
						7·33	151	1·681	4·35		1·10	0·00133		37·5	239·0	
—65	80	35	2410	—145	1·10					17·99			0·00541			1227·5
						7·33	138	1·540	4·76		1·10	0·00146		32·5	226·8	
—63	68	30	2070	—131	1·10					22·75			0·00687			1454·3
						7·33	125	1·392	5·26		1·10	0·00161		27·5	212·3	
—60	58	25	1725	—118	1·10					28·01			0·00848			1666·6
						7·33	112	1·247	5·88		1·10	0·00180		22·5	193·7	
—55	51	20	1380	—106	1·10					33·89			0·01028			1860·3
						7·33	97	1·080	6·78		1·10	0·00207		17·5	174·0	
—43	45	15	1035	—88	1·10					40·67			0·01235			2034·3
						7·33	76	0·847	8·65		1·05	0·00252		12·5	158·6	
—23	41	10	690	—64	1·00					49·32			0·01487			2192·9
						4·11	53	0·590	6·97		0·95	0·00184		8·6	88·0	
0	39	7·2	500	—39	0·90					56·29			0·01671			2280·9

appropriate constants for 4th gear, the average acceleration during the speed changes has been calculated and is tabulated in column 9. The time taken for each speed change has been calculated from the average acceleration and column 11 indicates the summation of these times, thus giving the total time required to accelerate from 7·2 to 60 m.p.h., or between any two intermediate speeds. In a similar manner the total distance travelled during the acceleration period has been estimated from the average speed and the distance covered during each speed change; and the total fuel consumption has been found from the average fuel consumption (obtained from Fig. 9) and the time taken for each speed change.

Thus, from Table I, it is estimated that to accelerate from 7·2 to 60 m.p.h. in 4th gear using the maximum torque conditions, takes 75·89 sec over a distance of 4758·9 ft and requires 0·2813 lb of fuel.

Table II was computed in a similar manner to Table I and indicates the method of estimation for a deceleration in 4th gear with the throttle closed, reducing the road speed from 60 to 7·2 m.p.h. Such a retardation would occupy 56·29 sec covering 2280·9 ft and using 0·01671 lb of fuel.

Tables I and II are typical examples of the tables used in calculating the acceleration and deceleration characteristics of the fourteen cases considered (see key). The results obtained from these tables are plotted in Figs. 10 to 15.

Figs. 10 to 14 indicate the eleven cases of acceleration considered. These curves, plotted on a time basis, show distance, speed and fuel consumption over the possible speed ranges attainable in the various gears. It will be noted that in cases where the throttle opening used is greater than that required to propel the vehicle at the maximum speed considered, the speed/time curve has a definite slope at this maximum speed (A2a, A3a, etc.) but when the throttle opening is just sufficient to attain the maximum speed, the curve tends to a horizontal straight line (A5a, A3c, etc.). In all the acceleration cases considered the fuel/time and distance/time curves tend to lines of constant slope after approximately one third of the total time, indicating constant rates of fuel consumption (lb-hr) and acceleration during the greater part of the acceleration periods.

The deceleration characteristics, shown in Fig. 15, indicate the differences in time, distance and fuel consumption between using engine braking in 4th and 5th gears and when freewheeling. Although the rate of fuel consumption is less when freewheeling the total fuel consumption is greater than when in gear owing to the increased time and distance required to decelerate from 60 m.p.h.

Table III compares the use of different gears and throttle openings in effecting similar speed changes. The last two columns of this table indicate the average fuel consumption (m.p.g.) and the "acceleration index" for the various cases. The quantity defined here as the "acceleration index" has been calculated as the ratio of the total fuel consumption (gal) to the time taken for the speed change (hr); its units are thus g.p.h. The smaller the value of the index the more economical has been the acceleration, whereas the larger the index the greater has been the acceleration and the shorter the time taken. If the criterion of fuel consumption (m.p.g.) is preferred, the comparison between the various cases is clearly shown in the penultimate column.

Table IV shows a comparison of deceleration characteristics in 5th gear (overdrive), 4th gear (direct drive) and when freewheeling for speed reductions of 60 to 15 m.p.h., 50 to 15 m.p.h. and 40 to 10 m.p.h. The deceleration indices have been calculated as before but naturally have very much smaller values than the acceleration indices. Furthermore, as the rate of fuel consumption is constant during periods of overrun and freewheeling, the deceleration indices in such cases will always have fixed values regardless of the speed changes involved. The value of the deceleration index for freewheeling represents the most economical

TABLE III. ACCELERATION COMPARISONS

Key	Gear	Power	Speed Change (mph)	Time Taken (sec)	Distance Covered (ft)	Average Speed (mph)	Fuel Used (lb)	Average Consumption (mpg)	Index (ghp)
A4a	4	Max.	15 to 60	69·78	4659·4	45·5	0·2744	23·55	1·931
A5a	5	Max.	15 to 60	116·27	8041·0	47·1	0·4038	27·65	1·702
A4a	4	Max.	15 to 50	34·03	1721·4	34·5	0·1093	21·87	1·576
A4b	4	Low	15 to 50	81·35	4794·2	40·2	0·2199	30·30	1·327
A5a	5	Max.	15 to 50	49·52	2535·0	34·9	0·1328	26·50	1·315
A5b	5	Low	15 to 50	107·22	6345·0	40·3	0·2562	34·40	1·171
A3a	3	Max.	10 to 40	17·06	650·1	26·0	0·0602	15·03	1·729
A3b	3	Low	10 to 40	40·97	1837·3	30·6	0·1119	22·80	1·341
A4a	4	Max.	10 to 40	25·18	946·0	25·6	0·0620	21·18	1·211
A4c	4	Low	10 to 40	74·85	3438·3	31·3	0·1360	35·10	0·892
A2a	2	Max.	10 to 25	5·64	147·0	17·8	0·0215	9·48	1·872
A2b	2	Low	10 to 25	15·21	439·0	19·7	0·0389	15·69	1·255
A3a	3	Max.	10 to 25	7·73	198·4	17·5	0·0199	13·85	1·262
A3c	3	Low	10 to 25	30·49	895·0	20·2	0·0461	27·02	0·742
A4a	4	Max.	10 to 25	11·83	304·4	17·5	0·0206	20·52	0·856
A4d	4	Low	10 to 25	46·10	1355·9	20·1	0·0512	36·85	0·545

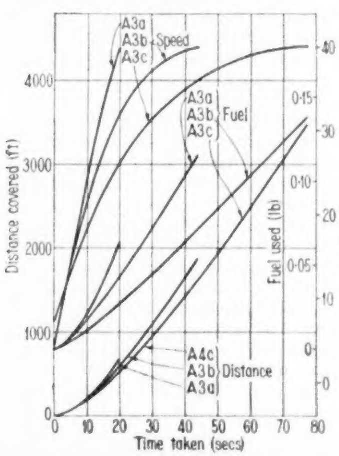


Fig. 12. Acceleration characteristics to 40 m.p.h.

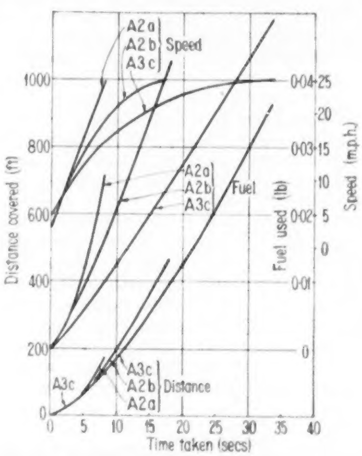


Fig. 13. Acceleration characteristics to 25 m.p.h.

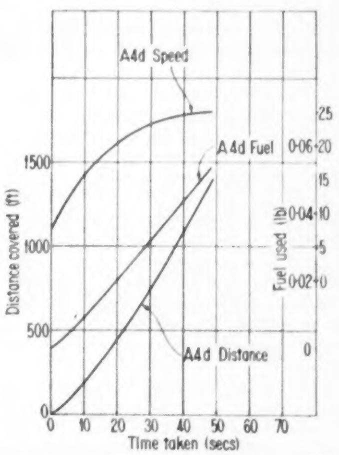


Fig. 14. Acceleration characteristics to 25 m.p.h.

TABLE IV. DECELERATION COMPARISONS

Key	Gear	Throttle Position	Speed Change (mph)	Time Taken (secs)	Distance Covered (ft)	Average Speed (mph)	Fuel Used (lb)	Average Consumption (mpg)	Index (ghp)
D4	4	Closed	60 to 15	40·67	2034	34·1	0·01235	228·5	0·150
D5	5	Closed	60 to 15	45·30	2232	33·6	0·01374	225·5	0·149
DO	Free-Wheel	Closed	60 to 15	69·51	3325	32·6	0·01738	265·8	0·123
D4	4	Closed	50 to 15	34·43	1540	30·4	0·01056	202·5	0·150
D5	5	Closed	50 to 15	38·88	1715	30·1	0·01183	201·3	0·149
DO	Free-Wheel	Closed	50 to 15	61·24	2660	29·6	0·01531	241·5	0·123
D4	4	Closed	40 to 10	35·68	1204	23·1	0·01079	154·9	0·150
D5	5	Closed	40 to 10	41·70	1400	22·9	0·01260	154·5	0·149
DO	Free-Wheel	Closed	40 to 10	64·33	2171	23·0	0·01609	187·5	0·123

manner in which a speed reduction may be accomplished, and consequently the most economical method in which any distance may be covered; this uses fuel at the average rate of 265·8 m.p.g. for the speed change of 60 to 15 m.p.h.

In order to produce a further comparison between the various cases of acceleration and deceleration considered, Figs. 16, 17 and 18 have been plotted showing the fuel consumption and time involved for complete journeys including periods of acceleration, constant speed and retardation. Fig. 16 assumes a journey of total distance 12,000 ft starting and ending at a speed of 15 m.p.h. and attaining a steady speed of 60 m.p.h. during the journey. One pair of curves indicates the manner in which the journey could be made in 4th gear, accelerating from 15 to 60 m.p.h. in 4,570 ft (A4a), running at constant speed for 5,396 ft and finally decelerating in 2,034 ft (D4). The other pair of curves shows the same journey completed in 5th gear, in which case the initial 8,041 ft are used for acceleration from 15 to 60 m.p.h. (A5a) and only 1,727 ft are covered at constant speed. In 4th gear the complete journey takes 172 sec compared with 177 sec in 5th gear, but 5th gear uses 0·502 lb of fuel compared with 0·594 lb in 4th gear, which is 19 per cent more.

Fig. 17 compares the vehicle performance in 4th gear with that in 2nd gear, changing to 4th when the steady speed has been attained. The journey distance in this case is taken as 2,000 ft starting and finishing at 10 m.p.h. and attaining a steady speed of 25 m.p.h. For the purposes of the comparison the gear change has been assumed to be instantaneous. The acceleration period in 4th gear covers a distance of 1,356 ft (A4d) compared with 147 ft in 2nd gear (A2a). The total journey time in 4th gear takes about 7 sec longer than when using a gear change, but there is a saving of 16 per cent in fuel consumption without a gear change. The acceleration in 2nd gear with maximum torque conditions has been taken as representing the greatest possible acceleration between 10 and 25 m.p.h. whereas the low-powered acceleration in 4th gear represents the most economical method of attaining this speed change. The intermediate use of 3rd gear on the journey would have produced a result between these two extreme cases.

Two different methods of decelerating the vehicle from 60 to 15 m.p.h. are compared in Fig. 18 over a distance of 4,000 ft. When travelling in 4th gear at 60 m.p.h., the throttle must be closed 2,034 ft before the end of the journey if the vehicle is to be left in gear, whereas if a freewheel is fitted this may be brought into operation 3,325 ft before the end of the journey and the engine left to idle at 500 r.p.m. The freewheeling produces a saving of 55 per cent in fuel consumption, but the time taken is about 14 sec longer than when retarding in 4th gear.

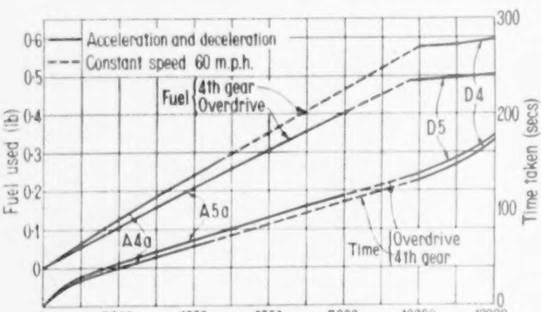


Fig. 16. Journey comparison between 4th gear and overdrive. 15 to 60 to 15 m.p.h.

Conclusions

The estimations clearly show the overall tendency that rapid acceleration can only be attained at the expense of fuel economy. Furthermore, to produce the best fuel economy the highest possible gear must be used at all times. It is interesting to note, however, that in order to cover even short distances, rapid acceleration in a low gear does not save an appreciable amount of time. For instance, from Fig. 17, accelerating from 10 m.p.h. in 2nd gear with maximum torque, a speed of 25 m.p.h. is attained in 5·5 sec over a distance of 147 ft, whereas in 4th gear and 13·7 per cent throttle opening the same distance may be covered in 7·5 sec, attaining a speed of 14·5 m.p.h. In 4th gear, however, the fuel consumption would be 0·007 lb compared with 0·022 lb in 2nd gear—a saving of 68 per cent in fuel consumption for a loss in time of 2·0 sec. Similarly, for the complete journey of 2,000 ft the loss in time using 4th gear would be only 7 sec compared with a fuel saving of 16 per cent.

Next, it has been shown that the addition of a 5th gear overdrive to the normal 4-speed gear box will produce a noteworthy fuel economy, especially on long journeys. The comparison between 4th and 5th gears (Fig. 16) shows a possible fuel saving of 19 per cent over a distance of 12,000 ft with a loss in time of only 5 sec on a total time of 172 sec.

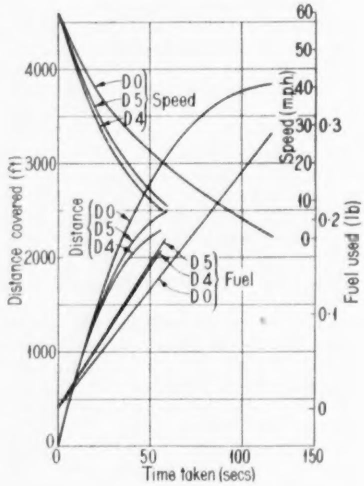


Fig. 15. Deceleration characteristics from 60 m.p.h.

The advantages of incorporating a freewheel device in the transmission system are illustrated in Fig. 18 where a fuel saving of 55 per cent is achieved in a distance of 4,000 ft. In this case, however, the increased journey time of 77 sec compared with 63 sec in 4th gear becomes more apparent.

Another interesting conclusion can be derived from the results. Fig. 16 shows that in 4th gear when accelerating from 15 to 60 m.p.h. and when running at a steady speed of 60 m.p.h. the slope of the fuel/distance curve remains almost constant. In other terms, the rate of fuel consumption (m.p.g.) during the acceleration period remains nearly constant and equal to the fuel consumption when the steady

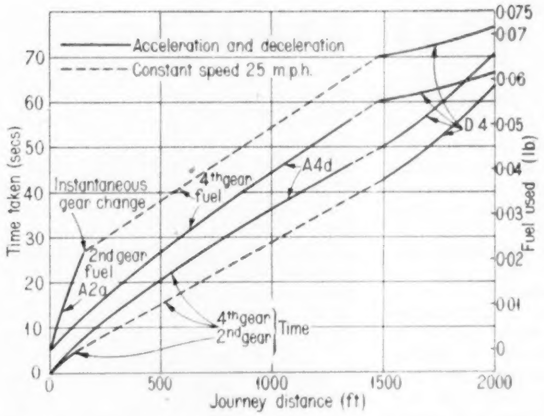


Fig. 17. Journey comparison between 4th gear and 2nd gear. 10 to 25 to 10 m.p.h.

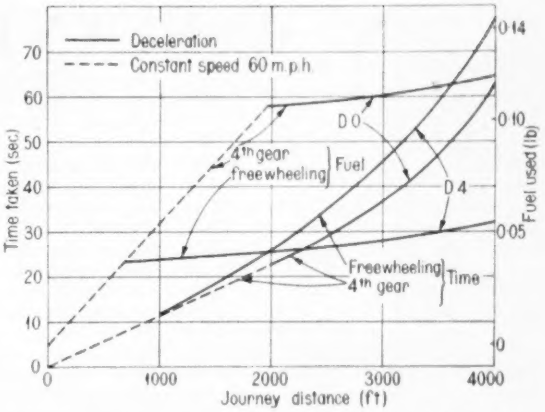


Fig. 18. Journey comparison between 4th gear and free wheeling. 60 to 15 m.p.h.

speed has been attained. Similarly, in 4th gear with 13·7 per cent throttle opening (Fig. 17), when accelerating from 10 to 25 m.p.h. the rate of fuel consumption remains sensibly constant and equal to the rate of fuel consumption at the final steady speed. Such curves show that, in general, the rate of fuel consumption on the level depends only upon the throttle setting and gear used, and is independent of vehicle speed. By careful calibration from engine test results, a scale could be arranged with a needle operated from the throttle control and the gear-shift mechanism, to read directly in m.p.g. Such a calibration would assume a level road and still air conditions and would only be accurate when the engine was accelerating the vehicle or propelling it at constant speed. The scale reading would be meaningless during periods of deceleration or when on a gradient.

It has been shown in a previous paper (Ref. 2) that, on a level road, the most economical speed of this particular vehicle is 32 m.p.h. giving a fuel consumption of 42 m.p.g. To maintain a steady speed of 32 m.p.h. on a level road would require a throttle opening of 18·7 per cent and therefore this setting would always produce a fuel consumption of 42 m.p.g. regardless of road speed. Consequently, to cover any distance with the maximum possible fuel economy a throttle stop should be arranged to restrict the maximum opening to 18·7 per cent. With the throttle against the stop the vehicle could be accelerated from 10 to 32 m.p.h. in 44 sec with a constant fuel consumption of 42 m.p.g. When the speed of 32 m.p.h. is attained the throttle would be closed and the vehicle allowed to freewheel, decelerating to 10 m.p.h. in 53 sec. During the total period of 97 sec the overall fuel consumption would be about 86 m.p.g., and the process could be repeated any number of times to cover a required distance. Although such driving could not be tolerated on a public highway, the method suggested represents the absolute maximum fuel economy attainable from the vehicle using the manufacturers' settings.

Finally, an application of equation (1) to the vehicle data in Arkus-Duntov's paper (Ref. 1) yields a curve for rolling resistance R_r, which is almost identical with his curve, obtained by a very involved process. This confirms the applicability of equation (1) even for speeds up to 150 m.p.h.

The experimental work required to obtain Figs. 1 and 2 was conducted in the Mechanical Engineering Laboratories of Queen Mary College, University of London.

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VIBRATION MEASUREMENTS

IN an article entitled "Vibration Measurements," by R. Winslade, in *Electronic Engineering,* December 1952, it is stated that a vibration detector can be sensitive to the displacement, velocity or acceleration of a vibration; the two quantities not measured directly can be obtained with an electronic differentiating or integrating network. Displacement detectors can be of the strain gauge, differential transformer or capacitance types. Acceleration detectors are also of the strain gauge or differential transformer types; both consist essentially of a mass mounted on a spring which permits motion in one plane only. Among the methods of electro-mechanical conversion used in velocity detectors are the movement of a coil perpendicular to a field of constant strength, or the movement of a ferromagnetic body relative to a magnet with a coil wound on it.

A simple differentiating and integrating network is described. A similar network is incorporated in an electronic calibrator, the function of which is to convert the "velocity voltage" produced by the pick-up to a voltage proportional to displacement, velocity or acceleration, and to measure the voltage so obtained, by comparing it with one of known value. Frequency of vibration can be measured by feeding a suitable frequency into the calibrator and comparing the wave form of the known frequency with the oscillogram produced by the vibration.

Two pick-ups are described: a seismic one, comprising two coils suspended on spring membranes and moving relative to a permanent magnet fastened to the housing; and an electro-mechanical one, in which the contact pin has a coil attached moving within the field of a permanent magnet.

The seismic pick-up is suitable for the measurement of very weak vibrations. Electro-mechanical pick-ups are intended primarily for very accurate measurements of relative oscillatory movements between objects, but can also be used for the measurement of absolute vibrations or for generating vibrations. The properties of the pick-up when used as a vibration exciter are analyzed, and the author shows that, in this capacity, it can follow only about 90 per cent of the acceleration. (*M.I.R.A. Abstract No.* 6246.)

NEW PLANT AND TOOLS

Recent Developments in Production Equipment

SINCE it will allow all the operations necessary to complete blanking and forming punches to be carried out at one set-up, the Thiel duplex 158 punch and die milling machine illustrated in Fig. 1 will be an invaluable adjunct in many toolrooms and die-making shops. It is of such proportions as to permit even hard materials to be machined at high rates of metal removal. The work table will support up to 300 lb weight. It has great rigidity, which is in great measure due to an inclined slide face that tends to counteract any deflection caused by vertical thrust. The universal table is arranged to swivel about both the horizontal and vertical axes. In addition, a sub-table can be supplied. It can be swivelled through 10 deg to either side of the longitudinal table axis. When used in conjunction with the standard table it allows fully universal setting.

Complicated control and drive mechanisms have been avoided. A new type of speed and feed selector dial is used. There is a choice of 12 speeds for the horizontal and vertical milling spindles, and eight speeds can be chosen, independent of the spindle speeds. The selector dials consist of three superimposed cones, each carrying a set of numbers. A table, attached to the machine, shows the speeds and feeds available and the possible combinations.

A sliding commutator motor supplies the main drive. This motor incorporates a highly efficient brake that is applied by sliding the commutator on to a friction cone when the motor circuit is broken. The braking is so efficient as to allow the tools to be changed at either the horizontal or vertical spindle without there being need for other clamping means. A touch on the brake release button immediately removes the brake. This button also serves for "inching" the machine.

Power feeds are provided for the three movements, and dual engagement levers are incorporated for the longitudinal table traverse. The length of travel for all three movements can be limited by trip dogs when the machine is used with power feeds, or by working to fixed stops when manual feeds are employed. In addition to the headstock traverse of 8 in, the overarm can be moved out to give a maximum throat distance of 29 in from the vertical milling spindle to the machine face. This wide range is very useful in machining large pieces of difficult shape for which re-location would be complicated and slow.

The vertical drilling and milling head greatly extends the scope of the machine. It is arranged to swivel through 360 deg. Efficient quill clamping is provided and the quill can be traversed by hand lever. A slotting head that swivels through 360 deg is also available. It has stroke and position adjustments. Either the drilling and milling head or the slotting head can be left in position while the machine is used for horizontal milling.

A universal dividing head, for direct and indirect indexing is available. It can be used horizontally or vertically. This head is arranged to swivel through 360 deg. It has a collet attachment, an outrigger steady, a tailstock, a face plate and three index plates. For milling accurate radii and other profiles a circular table for direct and indirect indexing, and complete with an auxiliary steady, can also be provided.

All three movements of the machine have slip gauge platforms so that the machine can be used for jig boring by means of the co-ordinate system. At the end of each gauge block platform there is a dial indicator holder. The slides also have scales and verniers and large calibrated dials to permit easy and accurate setting. All zero positions of work table and attachments can be readily located by taper dowel pins. An efficient coolant system is built into the machine. Rockwell Machine Tool Co. Ltd., Welsh Harp, Edgware Road, London, N.W.2, are the British agents for these machines.

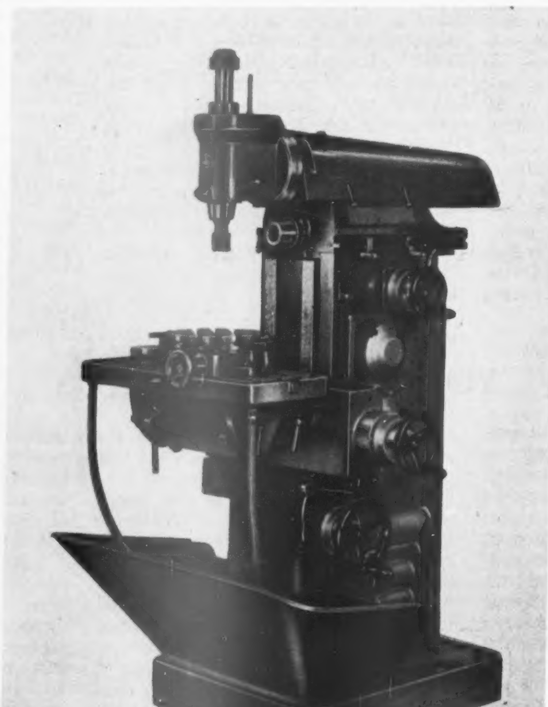


Fig. 1. Thiel duplex 158 punch and die milling machine
Rockwell Machine Tool Co. Ltd.

Rausch vertical broaching machine

The Rausch vertical broaching machine shown in Fig. 2 is an efficient but inexpensive tool for producing keyways, serrations and profiles, or for sizing bores on a wide range of components. The motor is flange-mounted on the rear wall of the column and transmits power to a coupling system located in a totally-enclosed, oil-filled gear casing. Two strong, square-threaded screw spindles extend in a vertical direction from the gear casing and are connected by a bridge with threaded sleeves of high-quality special bronze. The self-centring gripping jaws for gripping the broaches are incorporated in the bridge.

In both end positions the mechanical part is automatically disengaged through the coupling system. The return speed is twice that of the cutting speed. All the transmission parts are carried in ball or needle roller bearings, and end thrust is taken by ball thrust bearings. The table top has diagonally extending tee slots for mounting special fixtures for external broaching and an annular groove for accepting the accurate indexing attachment which can be supplied as an extra. In addition, there is a central bore for centring the expanding mandrel. A built-in ammeter serves for controlling the load on the broaching tools. There are two models, which differ only in length of stroke. One has a maximum stroke of 19¾ in and the other 35½ in. The pulling capacity is 17,600 lb and the cutting speed 4·9 ft per minute. The sole agents are Leo C. Steinle Limited, and the sole distributors in Great Britain are Burton, Griffiths and Co. Ltd., both of Marston Green, Birmingham.

Pacera drilling machine

As it has been developed primarily as a continuous service heavy duty machine, the latest addition to the

Fig. 2. Rausch vertical broaching machine
Leo C. Steinle Ltd.

range of "Pacera" drilling machines manufactured by W. J. Meddings Ltd., Slough Estates House, 16, Berkeley Street, London, W.1, is of exceptionally heavy and robust construction. It is illustrated in Fig. 3. This machine has a No. 2 Morse taper spindle and is rated at $\frac{5}{8}$ in in mild steel for continuous duty. The steel column is solid, is $3\frac{1}{2}$ in diameter and has a throat of $8\frac{3}{4}$ in.

An attractive feature of this machine is the wide range of speeds from 130 to 4,000 r.p.m. in 10 steps in both forward and reverse directions. A two-speed motor is employed so that it is possible to run at either of two speeds without changing the driving belt. All the table surfaces are ground and the bores in the head are honed. A neat table raising mechanism is fitted as part of the standard equipment. Other refinements include a graduated table to facilitate setting of the tilting table, a positive and newly designed depth stop mechanism, and convenient positioning of all the controls. A floor mounting model with a heavy cast iron bolster is also available.

Wolf electric saw

A 7 in heavy-duty portable electric saw, see Fig. 4, has been added to the range of tools designed and manufactured by Wolf Electric Tools Ltd., Pioneer Works, Hanger Lane, Ealing, London, W.5. It is known as the RS7 and has an extraordinary power/weight ratio. The motor is mounted transversely at right angles to the blade, with heavy helical gear drive. Thus, in action the saw rests in a balanced position on the main piece of the work to give greater safety, less effort and a clean cut.

All moving parts are dynamically balanced to give smooth operation, easy guiding and feeding. The saw blade is completely guarded. The depth of cut is adjustable to a maximum vertical cut of $2\frac{1}{16}$ in. Bevel cuts can be made up to an angle of 45 deg and to a depth of 2 in. A conveniently placed pointer allows angle adjustments to be made quickly and accurately. An adjustable ripping guide is provided as an item of standard equipment. An interesting feature is the introduction of a volute on the inner surface of the guard, which causes the sawdust to be ejected at the rear of the machine behind the operator. As a result the cutting line is kept clean.

A combined rip and cross cut blade of Sheffield crucible steel is supplied with each machine. Other types of blades are also available. These include a planer blade for a very fine finish, a fine-tooth blade for clean and fast cutting of wallboard and composition board, and a friction blade for cutting plain and corrugated sheet metal up to 20 gauge.

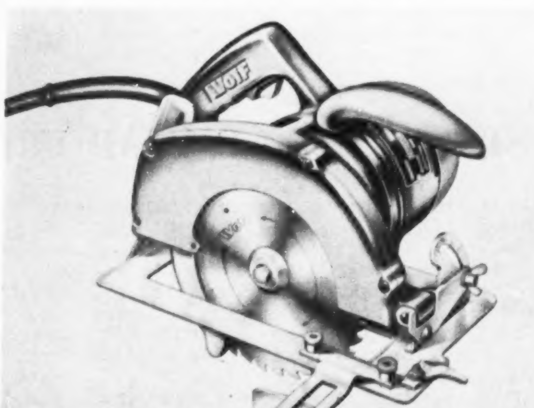


Fig. 4. 7 in heavy duty portable saw
Wolf Electric Tools Ltd.

Matrix broach sharpening machine

A machine that will be of great interest to the automobile industry has recently been developed by Coventry Gauge and Tool Co. Ltd. It is the Matrix No. 54 broach sharpening machine and is available in three models, one manually operated for round broaches only, one manually operated for both round and flat broaches and one for both round and flat broaches but fitted with sensitive pedal controlled hydraulic power cross movement. These notes describe only the model with hydraulic operation; the other models follow it closely in general construction.

The Matrix hydraulically operated machine, shown in Fig. 5, has been specially designed for sharpening all types of circular, flat and spiral broaches. It is of simple, straightforward design and is extremely accurate and reliable. An important factor is that no particular skill is needed to operate the machine. Briefly, the machine comprises a substantial base on which is carried a work slide that is actuated by means of a large handwheel conveniently placed in relation to the operator's position. Fixtures or a magnetic chuck can be mounted on the slide to take flat broaches. For round or spiral broaches a headstock and tailstock with steadies are bolted to the slide in any convenient position.

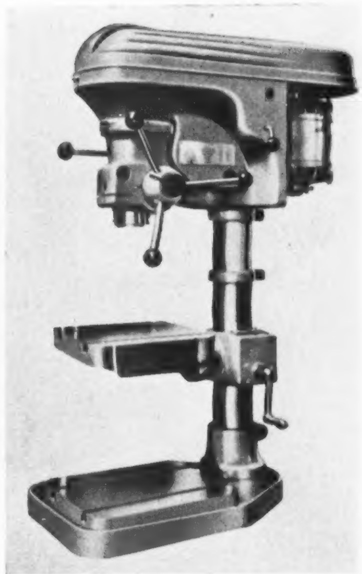


Fig. 3. Pacera HB2 drilling machine
W. J. Meddings Ltd.

At the rear of the base there is a specially balanced vertical slide that is actuated by means of a second large handwheel. This vertical slide carries the wheelhead unit. To accommodate spiral or shear cut flat broaches, the wheel head unit can be swivelled in a horizontal plane. The grinding wheel spindle is integral with its motor and can be inclined through any angle to suit the broach to be ground. This wheelhead spindle is clamped to a horizontal wheel slide that can be traversed hydraulically for flat broaches. The traverse is controlled by a pedal in the base recess.

The base has a knee hole so that the operator can be seated yet still view the work and the wheel without strain. Three sets of push buttons in the middle of the base control the grinding wheel, the hydraulic motor for supplying oil for the automatic traverse, and the work on the headstock. In the lower right-hand corner of the knee hole

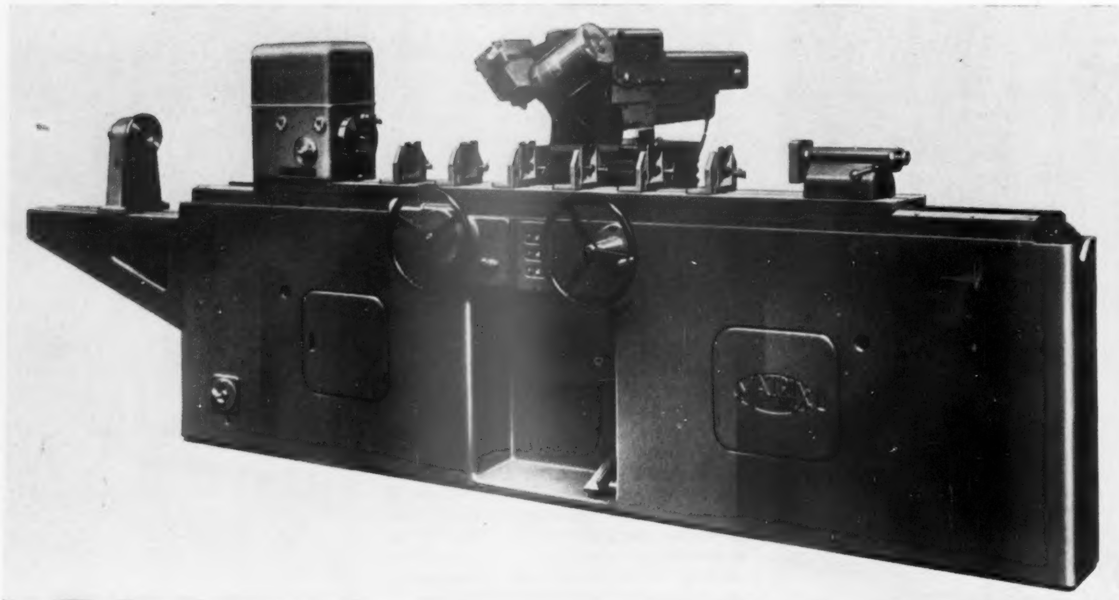


Fig. 5. Matrix broach sharpening machine
Rockwell Machine Tool Co. Ltd.

there is a pedal for starting the wheel slide motion and for controlling its speed when a flat broach is being ground.

The headstock is a self-contained unit clamped to the work slide by a single bolt for easy mounting and removal. A 0·6 h.p. motor drives the spindle through a belt drive that gives work speeds of 250 and 500 r.p.m. The driving plate is 7½ in diameter, is notched and has an indexing plunger for 2, 3, 4 and 6 divisions. This allows broaches having two or more sides to be sharpened between centres. A squared shaft projects from the front of the headstock so that a hand lever can be mounted to rotate the work spindle; this is essential for grinding certain types of spiral broaches. The tailstock is also secured by a single bolt.

For smooth operation the wheelhead slide is mounted on rollers. It is actuated by a simple hydraulic movement supplied through a pump and motor in the vertical slide. The wheelhead slide can be swivelled on its vertical slide to an angle of 45 deg for grinding shear cut broaches. The automatic traverse, controlled by stops, has a variable stroke up to 4 in. It can be disposed in varying positions to allow the wheel to grind centrally each side of the work table for shear cut and ordinary flat broaches. When a round broach is being ground, the hydraulic motor is switched off and the slide moved centrally until the zero marks on each side coincide.

The main wheel spindle is driven at 2,800 r.p.m. by a self-contained, 0·8 h.p. motor. This spindle unit is clamped to the horizontal slide by a single bolt and can be swivelled to any required angle. It is designed to take a wheel 8 in diameter × ½ in wide × 1¼ in bore. There are also adaptors for wheels 2 in × ¼ in × ⅜ in and 4 in × ½ in × ½ in. It is necessary to use a small wheel for grinding very small broaches and for such work a high speed spindle unit is supplied. This high speed spindle runs at 8,000 r.p.m. By means of an adaptor it will take a 2 in × ¼ in × ¾ in wheel, or a 1 in × ¼ in × ¼ in wheel can be screwed directly to the spindle nose. A profile wheel dresser is provided for dressing 8 in. diameter wheels. Smaller wheels are dressed by hand. The machine will accommodate circular broaches up to 4 in diameter and will admit 72 in between centres. Rockwell Machine Tool Co. Ltd., Welsh Harp, Edgware Road, London, N.W.2, are the sole agents for these machines.

FATIGUE STRENGTH AND LOAD FREQUENCY

A REPORT entitled "Influence of Test-Frequency on the Fatigue Strength of Steels and Light Alloys," by T. Wyss, is published in the *A.S.T.M. Bulletin*, February 1953. The investigation was made at the Federal Materials Testing and Research Institute in Zurich, Switzerland, following the development of high speed fatigue testing equipment. The materials tested included eight steels conforming to Swiss specifications and four aluminium alloys, the chemical compositions of which are shown in tables.

Static tests showed the steels to vary in modulus of elasticity between 27,500,000 and 29,600,000 lb/in^2 and in tensile strength between 67,500 and 142,200 lb/in^2. For low-speed tests, an Amsler 10-ton hydraulic universal testing machine of the packing-less lapped-piston type was used. The high-speed tests were done with an Amsler Vibrophore, which is electronically controlled and has a speed range of 100-300 cycles/sec.

Fatigue strength was determined under fluctuating tensile stress for 10^6 cycles. Both steels and light alloys were tested at a low frequency, 350 cycles/min, and at high frequencies of 10,500 cycles/min for steels and 8,000 cycles/min for light alloys.

At the higher testing frequency, low carbon steels show up to 7 per cent higher fatigue strength, and alloy steels up to 3 per cent. Light alloys are little affected by the testing frequency, though Perunal, an alloy containing zinc, shows a difference of 3·1 per cent. Actual differences are thought to be somewhat below those given by the numerical values of the tests. The use of high-speed fatigue-testing methods considerably reduces the time of testing; under 2 hr is needed at 10,500 cycles/min for 10^6 cycles, against 48 hr at 350 cycles/min. High testing speed also facilitates low and high temperature tests. (*M.I.R.A. Abstract No.* 6300.)

CYLINDER LINER DESIGN

Part I: Wear and its Causes

TO begin an article on cylinder liner design with a section on wear may at first sight appear somewhat incongruous. However, it is almost impossible not to do so, since the design problem is principally one of avoiding cylinder wear. It might be said that cooling is of equal, if not greater importance, but a little thought will show that the main object of cooling the cylinders, as distinct from the combustion chambers, is to reduce wear.

This is done, of course, by restricting the wall temperature to a level at which the physical properties of the materials of construction are adequate to cope with the loads imposed on them, and at which the lubricating oil will not deteriorate too rapidly. Cooling, embracing as it does cylinder heads and valves, as well as cylinder walls, is a subject that should be dealt with separately. This becomes the more apparent when it is realized that in a well designed engine, the heat flow through the cylinder walls is only a small proportion of the total heat lost to the coolant, most of which is transferred through the combustion chamber walls and exhaust ports in the cylinder head.

The wear problem

From time to time, it is pointed out that cars manufactured in the early 1920s suffered remarkably little from bore wear, and the question is asked why cannot we design engines nowadays to perform equally well in this respect? Unfortunately, there is no magic, known only to the craftsmen of that period, that could be applied to modern engines to cure or alleviate our wear problems.

The relative immunity of early engines from bore wear was due to a number of features inherent in the design not only of the engines but also of the vehicles. Lubrication of cylinder walls is by splash, whether from pressure fed big ends or from oil swept up by big ends in an all splash lubrication system. It follows that over a small part of the range of operating speeds, the amount of oil thrown up into the bores is ideal. At lower speeds, too little is supplied, and at higher speeds too much. In early engines, the operating speed range was much smaller than in modern ones, and it was therefore possible to arrange for optimum lubrication over a greater proportion of the range.

This effect should not be confused with variation of bore wear with engine speed. In fact, provided lubrication and cooling are adequate, and loads are restricted to a reasonable value, bore wear per stroke appears to be entirely independent of piston speed within practical limits.

Other conditions also influence the amount of wear experienced in modern cars as compared with those of the early 1920s. One is that modern cars generally are capable of much greater acceleration than the earlier ones. Moreover, because of the increased tempo of present-day traffic and the congestion on the roads, the accelerating capacity is generally used to the full. Even the improvements in brakes have had a marked effect on engine cylinder wear, since these improvements have been contributory factors making possible the increased performance already mentioned.

There is probably only one feature directly connected with the cylinder itself that could be said to have lead to the increased bore wear of to-day. That is the material used for the manufacture of cylinder blocks. The cast irons used then were in many cases better from the point of view of wear resistance than those now used. There are, however, several reasons why we cannot revert to the use of those materials. The close cylinder spacing and relatively thin sections used in modern engines are possible only with cast irons that flow freely in the casting stage. Moreover, these irons have been developed to produce castings entirely free from blow-holes and other defects that cause rejects.

If the better wearing irons were used, it would be necessary to increase the size of the block, and make the whole engine heavier and larger. This in turn would call for a bigger vehicle to carry it. Moreover, the number of rejected castings would be greater. The increased costs, arising from the use of such materials, would be of such an order that it would be preferable to fit cylinder liners which may be of even better materials. This measure is, of course, adopted in most commercial vehicles and also in a number of private car engines.

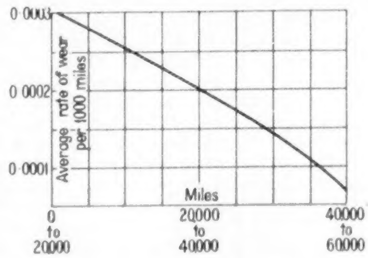


Fig. 1. The rate of wear decreases as mileage increases

Some wear is unavoidable, but by the use of suitable materials it can be reduced to negligible proportions. The rate of wear tends to decrease as the mileage is increased, Fig. 1. Generally speaking, when the amount of wear is about 0·25 per cent of the bore, oil control is affected, oil consumption will begin to rise, and in compression ignition engines, cold starting may become difficult because of loss of compression. When the wear is about 0·3 per cent of the bore, oil consumption will be excessive, blow-by will become noticeable, the piston temperature will rise and possibly may cause sticking of the rings in their grooves, and in the case of spark-ignition engines, the plugs will usually become oiled up after only a short period of running. As the wear increases to 0·4 per cent, the symptoms already mentioned will become worse, there will be a pronounced deterioration in performance, and the piston rings may break if they are still free.

The causes of wear

Much controversy has been raised on the subject of the causes of cylinder wear. This is because there are nine or more; in certain circumstances one only of these causes may be the critical factor, and under different operating conditions it may be another cause that gives rise to the trouble. For instance, under certain operating conditions overseas, the fitting of an air intake filter may have a marked effect in reducing wear, whereas this modification in the same country but on vehicles working under different conditions and frequently starting from cold may have little or no beneficial effect because the wear in this case may be mainly due to corrosion.

The main causes of wear may be listed under the following headings:

1. Excessive ring pressures.
2. Metal to metal contact causing rubbing wear.
3. Unsuitable materials for cylinders and rings.
4. Defective lubrication.
5. Corrosion.
6. Abrasive wear due to particles in the air or oil.
7. Detonation.
8. Excessively high temperatures.
9. High spots and distortion.

Before elaborating on these causes it will be necessary to describe in detail the mechanical conditions and changes that occur as the piston moves up and down in the cylinder. When the piston is in full flight, the rings ride up on the oil film and the conditions are those of complete fluid lubrication, but as the piston comes to rest at the ends of the strokes, the rings may be forced through the oil film and boundary lubrication results. That is, the two surfaces will be separated by an oil film of little more than molecular thickness, and there will probably be mechanical contact between the high spots on the rings and the cylinder.

It is the rings and not the piston that cause the greatest part of the wear. This is borne out by the fact that practically all the wear takes place on the part of the bore which is swept by the rings. At the bottom of the stroke there is little or no wear on that portion of the cylinder wall which is in contact only with the piston skirt. This effect is even more marked at the top of the stroke where, in a worn cylinder, a distinct step is generally formed on the wall at the level of the top face of the upper compression ring. Below the step, wear is excessive for a short distance, and above it there is little or no wear, Fig. 2.

Ring pressures

There are two schools of thought on the subject of compression ring pressures. In one it is held that the resilience of the ring should be sufficient to form the seal against the bore and that if gas pressure is allowed to get behind the ring it will add unduly to the radial pressure between the bearing faces and increase the wear. This view might be justified on the grounds that the wear problem would not be nearly so acute if the amount of metal removed at the top of the stroke were reduced to such an extent that wear became uniform over the length of the stroke. Wear at the top might be reduced by incorporating very small clearances between the rings and their grooves, thus restricting gas flow into the groove so that the pressure will not have time to build up behind the ring until the piston is well on its way down the bore.

The other viewpoint is that the main function of the compression rings is to form a gas seal, and, therefore, the radial pressure should be approximately proportional to the gas pressure. In fact, it should be slightly more than the gas pressure, otherwise the ring will collapse inwards and cease to function as a seal. The additional pressure required to make it higher is, of course, supplied by the resilience of the ring.

The Dykes ring is one that has been specially designed to ensure that the gas pressure gets into the groove to ensure adequate sealing between the ring and the bore. The theory put forward concerning the action of this ring is that, near the top of the stroke, a normal rectangular section ring is thrown against the upper face of its groove and prevents gas from getting behind the ring. When the pressure is high, on the firing stroke, the gas is forced between the bearing face of the ring and the bore and causes the ring to collapse inwards. To counter this effect Prof. Dykes designed an L-section ring to fit in an L-section groove. The clearances between top and bottom faces of the foot of the L and the groove are appreciably less than that between the top of the L and the groove. This means that the gas may get behind the vertical section of the ring and force it against the bore to effect a satisfactory seal. With this ring, the radial pressure during the idle strokes is considerably less than with a normal ring so that wear during these strokes is reduced.

Possibly, a good arrangement would be a compromise between the two schools of thought; a Dykes ring would appear to be well suited to act as a top compression ring, while normal rectangular section or taper faced rings might be used below. The reason why normal rings would be suitable for the second and third compression rings is that the gas pressure acting on the second ring is only that occasioned by the leakage past the top ring, and is usually little more than one-tenth of the combustion chamber pressure. The pressure on the third ring, if there is one, is even less.

Metal-to-metal contact

It is generally at the ends of the stroke that the greatest amount of rubbing wear takes place. The wear is most severe at the top end. This is partly because gas pressure builds up behind the top compression ring and adds to the effect of its natural resilience, forcing it through the oil film into closer contact with the cylinder wall. However, as will be shown later, this is not the only reason why wear is greatest at the top of the stroke; corrosion of this part of the cylinder is often the critical factor.

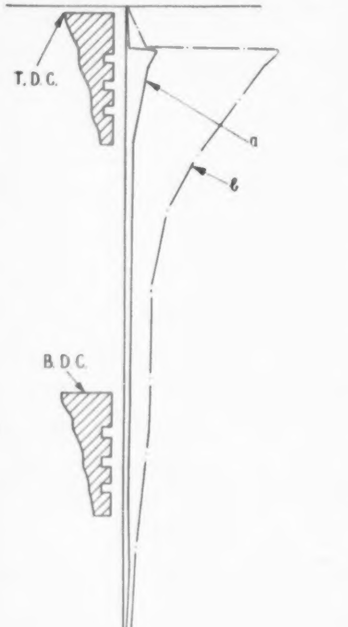


Fig. 2. Wear distribution over an 11 in long cylinder bore, (a) after 10,000 miles, (b) after 80,000 miles

When two metal surfaces are bearing together the area of contact between them is limited to that of the contacting peaks of surface roughness, or asperities. For example, it has been shown by the measurement of electrical resistance that steel flats under a load of 20 kgs and with an apparent area of contact of 21 cm^2 have a true area of contact of only 10^{-4} times the apparent area, F. B. Bowden and D. Tabor in *Proc. Royal Soc.*, 1939. Local pressure at the asperities must therefore be 10^4 times greater than the mean value expressed in terms of load over total area.

When the surfaces are sliding together the frictional work done must be dispersed as heat. In view of the relatively minute volume of each of the peaks in contact, it follows that their temperatures must be very high. This temperature is limited only by the melting point of the sliding metals. At temperatures approaching the melting point plastic deformation of the peaks may be expected, and at the melting point the peaks will flow. The effects obtained when lubricant is present are not measurably different from those obtained when it is not. This is because the hot-spots are formed so rapidly that the additional cooling agent has little effect.

At these high temperatures a metallic junction or welding occurs, and as the sliding motion continues these junctions must be sheared. The shearing may take place in any one of four different ways. If the junction is weaker than the metals themselves, shearing will occur at the interface. Consequently the amount of metal removed from either surface will be very small. This occurs, for example, with a tin base alloy sliding on steel. Junctions of this relatively weak nature are usually formed when tough films of oxide, sulphide or chloride are present on one or both of the surfaces. Even though these films are only of molecular dimensions they may, provided they are not broken up by deformation of the underlying metal, hinder the formation of strong metallic junctions.

If the junction is stronger than one of the metals, shearing will occur mainly within the bulk of the weaker metal, fragments of which will be left adhering to the harder surface. Under these conditions the amount of material thus removed may be relatively large, even though the friction is similar in degree to that observed in other cases in which little wear occurs. That is to say, there is no definite quantitative relationship between friction and wear. This type of wear results in the gradual build-up of a film of the softer metal on the harder one, so that ultimately the sliding is characteristic of that obtained with two surfaces of the softer metal. Consequently, the friction, surface damage, and wear are very high.

Sliding action between similar metals is the third way in which wear may take place. In this case, the process of

deformation and welding usually work-hardens the surfaces and appreciably increases their shear strength. Consequently, shearing rarely occurs at the interface, but takes place within the bulk of the metals. For this reason, the surface damage of both sliding bodies is considerable. On the other hand, a few alloys work-soften; these generally have good wear resistant properties.

The fourth type of wear is that which occurs when the junction is stronger than both of the metals. In this case most of the shearing will take place in the weaker metal, which will show the greater signs of wear. Nevertheless, it will also occur to some extent in the stronger one, in which the wear may still be appreciable.

Suitability of materials

From the explanation, just given, of the theory of wear it can be seen that hardness of the materials in contact is not the only criterion governing the rate of wear. Hardness is desirable for two reasons. In the first place, it tends to cause the shearing to take place at the welded face, since the mechanical strength of the material immediately below the welded surfaces is greater than would be the case with a softer material. To obtain this effect, it is only necessary for the hardness to extend to a depth of a few thousandths of an inch. Work hardening has a beneficial effect in increasing wear resistance. Secondly, if abrasives get between two hard surfaces, there is a greater chance of the particles being ground down to a harmless size than would be the case if the surfaces were soft. With unhardened surfaces, the particles may become permanently or temporarily embedded in one, and may then act as tools that cut grooves in, or score, the other surface.

In many respects melting point is a more important criterion than hardness. For instance, two contacting metals whose melting points are identical are likely to weld together at the asperities much more firmly than two with widely differing melting points. If their mechanical strengths are also equal, as is the case when the two surfaces are of the same material, then wear can be serious because of the work hardening effect on the surface.

This is the reason why, for instance, chromium plated rings should not be used in chromium plated bores. It also explains why, unless the material used for cylinder bores is carefully chosen so that its properties are suitable with respect to those of the rings, the wear resistance may be poor. Cast irons containing ferrite in their structure are particularly subject to welding or scuffing. On the other hand, the free graphite generally present in cast irons tends to inhibit welding so that a hardened and tempered ring may be used satisfactorily with a bore of similar material heat treated in the same way.

A chromium plated top compression ring is all that is necessary, and indeed all that is desirable. The lower rings should be of a suitable cast iron. When the relatively hard and highly polished chromium plated ring moves up and down the cylinder, it quickly burnishes over the asperities on the bore. This action is effected the more readily because the melting point of chromium is some 300 deg C higher than that of iron. The result of this burnishing action is that the bore beds to the ring instead of the ring to the bore. It is thought that the area of contact between the two is thus greatly increased, resulting in lower pressures on the asperities, lower temperatures, and therefore less welding and wear.

Since the amount of wear of the chromium is negligible, the bore can only bed to one ring, and if more than one chromium plated ring is fitted the result can only be increased bore wear because, as each ring slides over the cylinder wall in turn, it tends to change the contour to suit its own profile which is inevitably slightly different from that of the ring which preceded it. In fact, it can be argued that taper faced, cast iron compression rings should be fitted below the chromium top ring, since they will bed easily to the cylinder while causing a minimum amount of change to the bore contour which must eventually bed to the top ring profile. That such a marked reduction of wear is obtained by fitting a chromium plated top ring, tends to prove that the top ring causes the greatest proportion of the total wear. This is explained by the fact that the gas pressure behind the ring during most of the firing stroke is much greater than that behind the lower ones.

Lubrication

The effectiveness of lubrication is dependent not only on the properties of the oil, but also on the nature of the surface. In order to understand this it is necessary to study the interaction between the oil and the surface. In all substances, the constituent molecules or atoms are held together by strong cohesional forces between them. In liquids, the cohesion is generally sufficient only to prevent an appreciable quantity of molecules escaping from the surface to form vapour, but considerable translation and rotary motions go on within the liquid. On the other hand, in solids the molecules are held much more rigidly. In the interior of the material this molecular attraction is balanced on all sides. However, at the surface exposed to air there is an attraction inwards and to each side, but practically none outwards. When oil is spread over the clean surface of a metal, the surface molecules of the metal and of the oil are powerfully attracted to one another.

On a hone finished surface, some of the molecules of oil settle in to the valleys and crevices between the asperities and are held there by the attraction acting towards the centre of the metal and also on all sides. The attraction between these molecules and the main body of the oil film comes only from one side. Therefore, although the main oil film may be wiped off, the metal cannot be wiped dry easily because some of the molecules are firmly held in the interstices.

On the other hand, if the surface is burnished so that there are no interstices in which the oil molecules may be held, the attraction between the oil film and the metal is one of two plane surfaces together. Under these conditions the molecules of oil on the surface of the metal are subject to attraction from other oil molecules on all sides and towards the centre of the oil film. Since the oil, unlike the metal, is relatively free to flow, it tends to form pools and the spaces between the pools remain dry; in other words it does not wet the surface of the metal. Because of this tendency of the oil to leave dry patches on smooth surfaces, chromium plating, if used on cylinder bores, must be given a matt, or satin, finish.

Under conditions of effective lubrication the piston rings and cylinder bore are separated by a film of oil some hundreds of molecules thick and all the valleys between the asperities are filled with molecules. Provided there are no abrasives large enough to span the film, mechanical wear cannot occur because there is no solid contact between the rings and bore.

However, under heavy load, the main film may be squeezed out from between the two surfaces. The condition of boundary lubrication thus obtained is still satisfactory since the oil molecules adsorbed in the interstices still remain, and their attraction to the surface is so great that they maintain a surface film even though it is of infinitesimal thickness. These oil molecules are sufficient to separate the surfaces. The coefficient of friction is usually a minimum at the transition point between boundary conditions and full lubrication. This is probably because the attraction between the molecules of adjacent layers in a thick oil film is greater than that obtained with the broken film of the boundary lubrication condition.

In a paper entitled "Cylinder Wear, Where and Why," by S. W. Sparrow and T. A. Scherger in the *S.A.E. Transactions*, April 1936, an experiment is described in which the face of an oil control ring was filed away for a length of one inch on its circumference. It was then assembled to the piston with the filed away slot at the centre of the thrust face. The ring was pegged in its groove to prevent rotation. In addition, the compression rings were pegged in their grooves, and the cylinder wall was sand blasted to give an exceptionally rough finish and thus accelerate wear in the initial stages. A half tone illustration of the cylinder block, which was sectioned longitudinally after a one hour run at 3,000 r.p.m. on full load, shows clearly where the wear has taken place. Unworn areas retained their frosted finish, while the worn ones became polished.

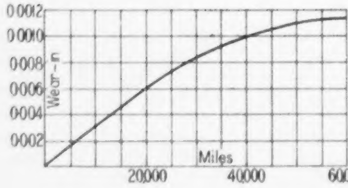


Fig. 3. Oil control becomes less effective as the bore wears, and in this vehicle, the resultant excessive lubrication has appreciably reduced the rate of wear after about 40,000 miles

The interesting feature of this experiment is that it demonstrated the fact that where the oil control ring was ineffective because of the slot in its face, the frosted surface remained. This occurred despite the fact that the compression rings would undoubtedly exert a greater than normal radial pressure on this area because it would project beyond the level of the surrounding worn areas of the bore and form a high spot.

Two conclusions may be drawn from this. Firstly, an over-effective oil control ring may be the cause of bore wear, and secondly, if oil control requirements can be relaxed, the compression rings will ride over the oil film and the resulting full fluid lubrication conditions will eliminate practically all rubbing wear. This effect is probably the main cause of the reduced rate of wear experienced as the bore diameter increases in service, Fig. 3.

Unfortunately, oil control cannot be relaxed as a design measure, since excessive oil consumption increases running costs and fouls sparking plugs. In addition, in many cases it is not necessary to increase the oil supply to the bores, because rubbing wear is not the critical factor. Corrosion might be reduced by furnishing better protection, in the form of extra oil, to the surfaces, but there is no evidence that such measures would have this effect in all engines.

So far as lubrication is concerned, viscosity is another factor that influences bore wear. When temperatures are high, the viscosity of the lubricant is reduced and the rings are more prone to break through the oil film and run under conditions of boundary lubrication. At these high temperatures, the more viscous oils should be used; for not only do they delay the break-through, but also when it does occur, boundary lubrication is effected more satisfactorily by the higher viscosity lubricants.

On the other hand, when starting from cold, S.A.E. 20 grade oils will reach the top ring in half the time taken by S.A.E. 30 grade. Therefore, the bore wear in vehicles operating under conditions where frequent starts from cold are made, may be reduced by using the thinner oils. Nevertheless, there is a tendency for the more stringent oil control, necessary to prevent excessive oil consumption with oils of this type, to offset the advantages obtained from their low viscosity. However, this tendency may be largely circumvented by using hardened and tempered oil control rings instead of the more usual as-cast ones. With hardened and tempered rings a sharp oil control edge is better maintained since scuffing is reduced as a result of the almost total elimination of free ferrite in the structure.

It is often forgotten that oils suitable for use in Great Britain are not necessarily to be recommended for use in overseas territories. In this country, because extremes of temperature are infrequently experienced, and cars are not usually left standing in the open for long periods, the effect on bore wear of the viscosity of the oil used in engines subject to normal usage is not so marked as in countries with less temperate climates.

In America climatic conditions are very different and cars are frequently parked in the open overnight. It appears that there, heavy oils are too viscous to reach the top ring quickly enough during cold conditions and the film strength of thin oils is not great enough to give effective lubrication at hot running temperatures. Tests have shown that lubricants having an intermediate viscosity rating are best for general use in that country. It must be emphasized that these remarks, as well as those relating to conditions in the British Isles, are mostly of a general nature and do not necessarily apply under all operating conditions. In addition, there are other properties of lubricants, besides viscosity, that may influence bore wear, but this subject is so complex that it could only be dealt with in a separate article.

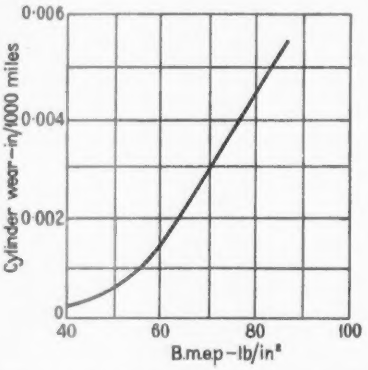


Fig. 4. At low temperatures bore wear increases rapidly with load

Experiments with upper cylinder lubricants have generally given disappointing results. This is not surprising when consideration is given to the fact that the proportion of upper cylinder lubricant in the fuel is necessarily small. It is easy to estimate from the fuel consumption figures how much lubricant is introduced into each cylinder during the first five minutes after starting from cold. Even when making a most conservative allowance for the amount of lubricant passing straight out with the exhaust and for the amount deposited on the cylinder head, the quantity of lubricant that can reach the critical areas of the cylinder walls will be found to be minute.

Nevertheless, a number of the proprietary upper cylinder lubricants are claimed to have other properties. For instance, some are of an alkaline nature to neutralize the acids formed during combustion. These acids certainly accelerate corrosion when they are deposited on the bores during cold running. Other proprietary brands contain colloidal graphite which is said to be deposited on the walls and to resist being washed off by any petrol introduced into the cylinder during an over-rich start from cold.

Corrosion

Petrol engines are particularly subject to the effects of corrosion because of the tendency of unskilled drivers to start from cold on an over-rich mixture. Under these conditions neat petrol may be drawn into the cylinders and wash the protective film of oil off the bores. Diesel engines, on the other hand, are also liable to corrosion because of the impurities in some of the grades of fuel oil used.

Sulphur is one of the most detrimental impurities in diesel fuels since after combustion it forms SO_2 and SO_3 which combine with the water of combustion to give sulphurous and sulphuric acid. During the combustion of various fuels, other corrosive substances developed in small quantities include carbonic acid, formic acid, acetic acid, nitric acid and, in leaded fuels, hydrobromic acid, and of course the water already mentioned.

When the engine is at its normal running temperature these detrimental products of combustion remain vaporized and are discharged through the exhaust system without having any serious effect on the cylinder bores. However, the dew point of many of these vapours is high under the conditions of pressure in an engine cylinder. In fact, to be safely above the dew point, the cylinder surface temperature should be raised above 120 deg C. as quickly as possible after a cold start. This is because corrosive vapours tend to condense on surfaces which are at a lower temperature. The rate at which an engine may be warmed up is, of course, materially increased if a thermostat valve is incorporated in the cooling system.

Experiments have shown that under cold running conditions most of the wear takes place at the top of the stroke. The reason why this is so may be deduced from an examination of what happens as the piston approaches and recedes from top dead centre. Not only is the gas pressure behind the first compression ring high at the top of the power stroke, but also the thrust changes from one side of the cylinder to the other. These conditions lead to the oil film, which is in any case very thin at this level, being squeezed from between the ring and the bore, and metal to metal contact results.

The rubbing action of the ring as it descends then tears minute pieces of metal from the surface of the bore. This leaves the cylinder wall in this area not only denuded of its protective lubricant, but also with the surface bright and rendered chemically active by virtue of the fact that there is no protective film of oxide on it. On successive strokes the products of corrosion are swept off, and may act as abrasives, further increasing the amount of wear. Under cold starting conditions the rate of wear increases as the load is increased, Fig. 4.

From this examination of the wear process, it can be seen that there are other measures besides rapid warming up that may be taken to alleviate the troubles. For instance, wear may be appreciably reduced by plating the bores with a corrosion-resistant material such as chromium. An additional advantage is obtained with this material: because of its hardness there is little or no tendency for minute particles to be torn from it by the rings so that the bare metal is not exposed to the corrosive influences. The rings do not generally suffer much from corrosive wear, since they are adequately supplied with lubricant. Chromium plating does not completely protect the bores, because it must have a porous surface to retain the oil. Therefore, there may be a slight tendency for acids to pass through the pores and attack the underlying ferrous metal.

Abrasion

Abrasives may get into the engine in the following ways:

1. Core sand, metal swarf and dust may get in during manufacture and may not be completely removed by the subsequent cleaning processes.
2. Valve grinding compound or cylinder honing residue may be introduced as a result of inadequate cleaning after servicing.
3. Under operating conditions, dust may enter the crankcase through the ventilation system.
4. Dust may enter through the air intake of the induction system.
5. Abrasives are formed by the corrosion and wear processes already described.
6. Some fuels form abrasive ash during combustion, particularly at abnormally high temperatures.

The greatest possible care is taken during manufacture to remove all abrasives, and the only other precaution that can be taken is to change the oil and flush out after the running-in period. Better control of servicing operations should eliminate trouble under the second heading of the list. With regard to the third, it may be said that the importance of fitting an air filter to the intake of a crankcase lubrication system is not always appreciated. Dust entering the crankcase is naturally picked up by the oil, and many consider that it causes much more damage to the engine than dust entering through the air intake.

However, although much of the dust breathed in through the induction system is undoubtedly carried straight out again through the exhaust, it is probably more detrimental than that in the oil so far as cylinder wear alone is concerned. For this reason, air filters are fitted to the air intakes of most modern vehicles. Most of the wear arising from dust entering through the induction system takes place at the top ring, not only on the part of the bore that it traverses, but also in the ring groove. By the time the abrasive material reaches the lower rings it generally has been so pulverized that it is relatively harmless. For, to do damage, the particles must be large enough to span the lubricating oil film, and this is thinnest at the top of the bore.

The products of the corrosion and rubbing wear processes often form fine abrasives, but their action is one of mild lapping rather than of scoring. At high temperatures certain grades of diesel oil tend to form an abrasive ash, and this may have damaging effects both on the bore and on the top ring and its groove. The hard carbon formed from burned lubricating oil on the top land also may have an abrasive effect.

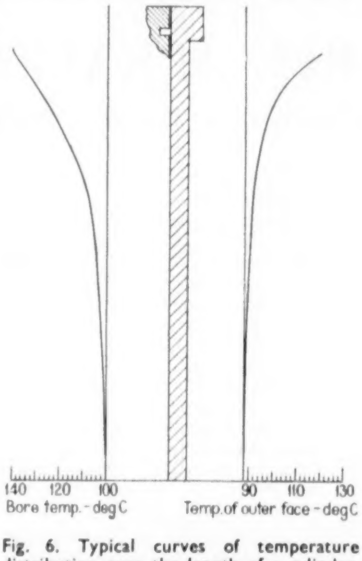


Fig. 6. Typical curves of temperature distribution over the length of a cylinder. The water temperature at the top was 90 deg C and at the bottom 77 deg C

Heavy scoring may cause blow-by which, if it is severe, may tend to scour off the lubricating film, burn the metal surfaces, and generally increase the temperature of the adjacent areas. When engines are operated under very dusty conditions, the larger particles may bed in the piston skirts and cause a considerable amount of scoring of the bores both above and below the area swept by the rings. Abrasion may be reduced by chromium plating the bores, or by using austenitic iron liners; however, these measures increase costs.

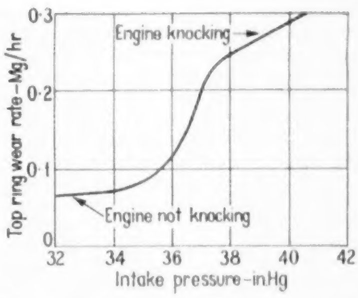


Fig. 5. A marked increase of wear is experienced when detonation occurs

Detonation, temperature effects and distortion

Detonation has been shown to increase bore wear, Fig. 5, and it is possible that variations in the wear of different cylinders of the same engine may be due to the fact that detonation has been more severe in some than in others. The primary effect of detonation is to subject the cylinder walls to severe mechanical stresses and to increase the heat flow to them. This increased heat flow is due to the scouring away from the walls of the stagnant boundary layer of gas which otherwise would act as an insulator. In addition, the heat is developed in the cylinder much more quickly than under normal conditions of combustion, so there is more time during the stroke for transfer to take place. Cylinder wall vibrations set up by detonation may tend to cause wear. Wet liners held in position only by a flange at the top end would appear to be particularly liable to vibrate.

The effect of high temperatures is to evaporate the lubricating oil film, or to burn it and form sludge and gum. The gum formation may in turn cause the rings to stick. Under these conditions extremley high pressures may occur locally at some points between the rings and the bores, and at other points excessive blow-by may be experienced. Both these conditions tend to raise the wall and piston temperatures even higher and cause increased wear. Another effect of high temperatures is the lowering of the viscosity of the oil, but this has been discussed already under the heading lubrication. It would appear that the optimum surface temperature of the liner at the point of maximum wear is about 140 deg C. A typical temperature distribution diagram is given in Fig. 6.

Another temperature effect that sometimes causes scuffing and seizure may occur on starting from cold. If an aluminium piston gets warm at a much faster rate than the cylinder, the running clearance may be taken up, with disastrous results. This effect is, of course, more likely to cause trouble when aluminium alloys with a high coefficient of expansion are used for the pistons, but the running clearance specified is generally adequate for all conditions.

Hot spots or uneven temperature

distribution is a more frequent cause of wear than general overheating. The effect of local overheating is to cause distortion, since the expansion at the hot spot is greater than that of its surroundings. This may cause bores to go out of shape so that the rings no longer fit properly and blow-by occurs, or it may cause a local high spot where excessive wear and scuffing are likely to take place. Another common cause of distortion is badly designed cylinder head holding-down arrangements, or uneven tightening of the holding-down nuts or bolts. The fact that wear occurs in different places in different cylinders of the same engine and is not necessarily greatest on the thrust faces may often be accounted for by distortion.

Symmetricality of design is essential to avoid distortion troubles. This is difficult in some designs, particularly in side valve engines. Severe changes in section should not be introduced, and the material should be arranged symmetrically around the cylinders. The holding-down stud arrangement should be such that the head is pulled down evenly, and the bosses into which the studs are screwed should be solidly attached to the block and preferably well clear of the cylinder walls. A good stud arrangement is that used in the S-type Bedfords in which long studs are employed, and they are screwed into bosses in the base of the cylinder block instead of into the joint face at the top, Fig. 7. Another important requirement is that the cooling flow be directed over the hottest parts where the most cooling is needed to maintain even temperature gradients throughout the structure. Stud bosses should not impede the flow or form pockets in which vapour may collect.

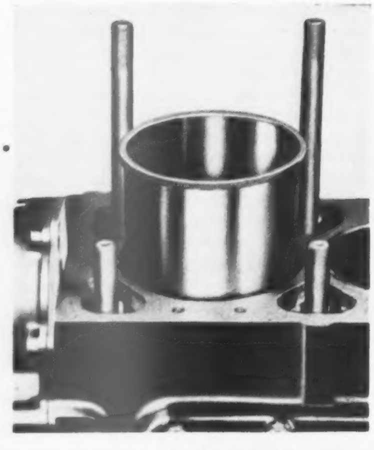


Fig. 7. The cylinder head holding down studs of the S-type Bedford are arranged to avoid distortion of the bore

(To be continued)

RECHARGEABLE ALKALINE BATTERY

An Interesting American Design

INFORMATION has been received from the United States of America concerning a new type of rechargeable alkaline electric storage battery which has been developed by The Alkaline Battery Corporation Inc., 520 Fifth Avenue, New York, U.S.A. It is said to be of a completely new design and construction that differs radically from all other alkaline batteries as well as from the more commonly used acid lead type battery.

The salient features of the battery may be summarized as follows:

(1) There is no lead used in the construction of this battery. It is of all steel construction with 25 times the structural strength of the lead type.

(2) In contrast to most batteries in common use there is no acid electrolyte used in the ABC alkaline battery. Instead it has a potash electrolyte which is actually a preservative of steel.

(3) Neither cadmium nor silver are used. It is strictly a nickel iron battery. This is important economically as well as technically, since at present cadmium in the form used costs approximately 2·40 dollars per lb of powder, whereas iron powder used in the ABC battery costs approximately 15 cents per lb.

(4) There are no fragile containers to crack or break. The conventional battery has a hard rubber or plastic container. A welded container is used for the ABC battery.

(5) As a laminated construction, placed under high hydraulic pressure during manufacture and so held for the life of the battery, is used, it is not possible for the active materials to shed and drop to the bottom of the case.

(6) As the plates are assembled in a stack under hydraulic pressure and positively locked by the end plates and locking rings, there is no possibility that they should buckle, with a consequent destruction from the short circuits.

(7) This battery has a very low internal resistance. This is important when it is considered that the battery may be normally charged in one hour without excessive rise in temperature and without any possibility of damage to the battery itself.

(8) It is impossible to damage the ABC battery by under or overcharging. In addition the battery can stand idle for indefinite periods of time and still hold its charge.

Test results

Compounds of iron and nickel are used for the active materials of the ABC battery and caustic potash is used as the electrolyte. The structural and current carrying parts are made from nickel-plated steel. At normal rate the average charge voltage is 1·58 per cell; the average discharge voltage is 1·30 per cell.

One battery made in 1946 was first given a five years' laboratory accelerated test and was then put into an automobile and has been in continuous service for five years. It has been given sufficient deliberate abuse to have destroyed several conventional batteries. For example, it has been allowed to run dry, without there being any noticeable effect on performance. It has been discharged completely and deliberately overcharged. This battery is still in daily service and is more than 90 per cent effective.

In a further test 10 cells were cycled on full charge at normal rate followed by full discharge at 10 times normal rate. After 1,000 cycles their five hour rate capacity was only 8·7 per cent below rating.

Another group of 10 cells was given an accelerated life test under the following conditions:—

5 seconds discharge at 30 amps.
3½ minutes idle.
6 min 25 sec charge at 10 amps.

This cycle was intended to simulate actual service in an automobile. The test ran for 24 hours a day and was interrupted every 500 cycles to allow capacity tests to be made. After 127,000 cycles the average capacity had fallen off 20 per cent, at which stage the test was discontinued. Without reference to the time element, this amount of work is estimated to equal a service life of 25 years in an automobile.

A 75 A.H. ABC cell has an internal resistance of 0·002 ohm when fully charged and discharging at a five hour rate. The internal resistance shows little increase as discharge progresses. This is shown by the fact that a 6·5 volt 65 A.H. ABC battery will yield over 90 per cent of its total capacity above 3·5 volt at a discharge rate of 300 amps. In comparison, a new 6·0 volt lead battery of 100 A.H. capacity gave only 52 per cent of its total capacity on the same test.

It is claimed that the ABC battery is simpler and cheaper than any other type of alkaline battery. For example, an Edison alkaline battery cell has a total of 820 parts whereas the ABC cell has only 200 parts.

INDUCTION HEATING

Some Recent Automobile Applications

D. Warburton Brown, A.M.I.Mech.E.

AFTER a decade of commercial development, high frequency induction heating has now reached the stage of being recognized by production engineers in the same light as other machine tools. It is no longer viewed with misgivings but its relative advantages are carefully assessed, and weighed from all aspects against other methods. The fact that there is now a thriving industry producing this type of equipment indicates that for many applications the process does have definite economic advantages.

The process has been applied successfully to a number of applications in the automobile industry involving soldering, brazing and hardening operations; in the following notes some of these are detailed to give automobile engineers some idea of the scope of the process. Naturally, times and methods of handling vary with the physical dimensions of the pieces to be treated but the examples given will indicate the order of production times and the principles utilized in handling the components. It should be borne in mind that with higher powers it is generally possible to obtain more rapid production times. It is not possible to give details of every possible application on a modern motor car, but it is hoped that the information given will enable automobile engineers to consider the possible application of induction heating to their own particular products.

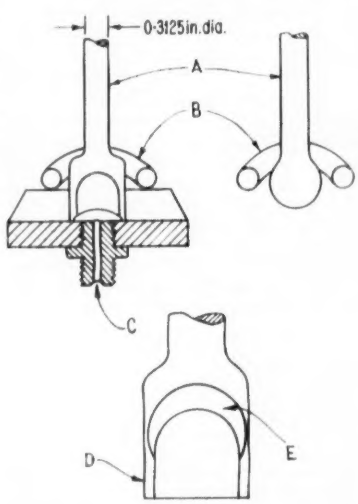
Push rods

The push rods used in many automobile engines employing overhead valves generally consist of a steel rod or tube terminating in a ball at one end and a spherical socket at the other. The surfaces subjected to wear are the extreme ends where the rod comes into contact with the tappet and with the rocker adjusting screw. To resist wear it is desirable that the extremities of the push rods should be hardened and this operation has been successfully carried out by induction heating.

The power required naturally depends upon the dimensions of the push rod being treated, but as an example the case of a push rod 8½ in long and having a stem diameter of ¼ in enlarged to a ⅜ in sphere and cup can be quoted. This push rod had the two ends hardened simultaneously in 12 sec on 2½ kW equipment. The rods were handled on an automatic fixture that fed them to the work coils which were arranged on hinges to enable them to be brought into the correct relationship with the work after it had been fed to the heating position.

In another case the manufacturer requested that the hardness in the cup end of the push rod should conform to the pattern indicated in Fig. 1. The reason for leaving the extreme edges of the cup in an unhardened state was that it had been found that owing to the thin section of the metal at this point breakages occurred in transit when the complete cup was hardened. These rods were treated in 5 sec on a 15 kW generator. In this case a single turn plate coil was used and quenching was carried out with the push rod *in situ* as indicated in Fig. 1.

A further application of induction heating to push rods can be found in the case of a manufacturer of heavy goods vehicles who designed push rods in the form of a high tensile steel tube with hardened ball and spherical socket soldered to the ends.



A. push rod. B. work coil. C. quenching water. D. edge left soft. E. hardened area.

Fig. 1. Arrangements for hardening push rods

In the original method the end components were first copper plated all over and a hole drilled in the barrel of each to permit them to be dipped in a bath of molten solder for tinning. The end fittings were then pressed over the tube and re-heated by gas torch while solder was applied to fill the joints. These joints were quite efficient, being capable of withstanding loads of approximately 2 tons, but a separate operation to remove surplus solder was necessary. A further difficulty with the original method of using a gas torch was that care had to be taken not to let down the hardness of the end fittings.

With a 4 kW induction heater the same operation was carried out using solder paint in conjunction with a ring of solder wire and it was possible to delete the copper plating operation and the drilling of the end pieces. The time taken for these joints was in the order of 5 sec for each end and tests indicated that the strength of the joints had been increased to nearly 3 tons/in². A simple rotary fixture with foot control indexing was used so that one assembly could be lowered at a time into the heating coil which consisted of 4¼ turns of $\frac{3}{16}$ in copper tubing.

Tappet screws

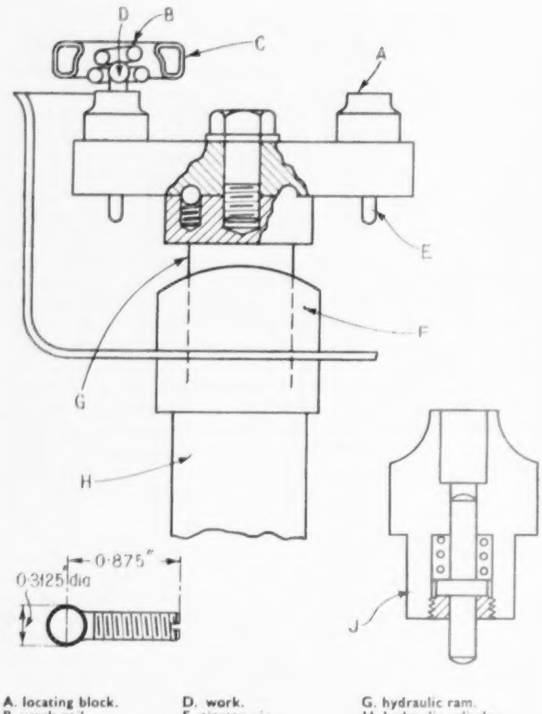
Associated with the valve gear on automobile engines is the small spherically headed adjusting screw which provides the means of adjusting tappet clearances. These screws have a spherical end that has to be hardened to resist wear due to rubbing in the sockets of the push rods. The hardening of the ball end has been satisfactorily carried out by induction heating with a work coil in the form of a cap fitting over the spherical screw. This type of coil is sometimes referred to as beehive coil.

As the physical dimension of the screw is only small, it is not necessary to use high powers and successful results have been obtained from equipments with outputs varying from 2½ to 6 kW, and production times have been obtained in the region of 3 to 7 sec per piece, depending upon the size of the screws and the power used. Hardness figures in the region of Rockwell C.60 are obtainable with oil quenching steels without any sign of surface or internal cracking of the metal.

Fig. 2 illustrates a very simple handling gear for dealing with the hardening operation on a production basis. Briefly the equipment consists of a rotating turntable with suitable locating blocks arranged around the periphery. This turntable is mounted on a foot-operated hydraulic ram which enables the components to be brought into correct relationship with the work coil and quench ring. Suitable provision is made for ejecting the finished pieces.

Rocker shafts

The shaft on which the rockers are mounted in overhead valve engines is generally made from steel tube or bar in ½ in to ¾ in diameter range. It is now fairly common practice to surface harden these shafts at those points where they are subjected to the rubbing action of the rocker issue. Generally this localized surface hardening operation can be satisfactorily carried out



A. locating block. D. work. G. hydraulic ram.
B. work coil. E. ejector pin. H. hydraulic cylinder.
C. quench ring. F. ejector cam. J. details of ejector block.

Fig. 2. Set-up for hardening tappet screws

by means of induction heating, and using an En 9 material which is a straight carbon steel having a carbon content of 0·55 per cent. Surface hardnesses in the region of Rockwell C.60 can be obtained and an average case depth of 0·040 in is obtainable.

In practice it is found more satisfactory to treat the hardening of these shafts on a progressive basis, that is the shaft is traversed through a single turn work coil and an adjacent water quench ring as shown in Fig. 3. The shaft is usually made to rotate during the traversing period. When the hardness is only required at local points, it is an easy matter to arrange for the power to be switched on and off at the appropriate points, this being a purely mechanical problem. Naturally the speed at which the hardening operation can be carried out will depend upon the diameter of the shaft and the graph given in Fig. 3 will enable the production time for any shaft to be estimated. The graph is based on test results using a 15 kW heater operating at a frequency of 500 kilocycles and the results plotted for an average case depth of 0·040 in. With a 15 kW equipment the practical maximum diameter of shaft is 1·5 in. To take an example, if it is required to case harden a shaft ¾ in diameter and 12 in long, the time required, as can be read from the graph, is 3½ sec per inch or a total of 42 sec.

Rocker pads

The working face of the rockers in overhead engines has to be hardened in order to withstand rubbing action of the tip of the valve and this hardening operation has been carried out with complete success using induction heating equipment with a power output of 6 kW.

On rockers made from straight carbon steel with a 0·45 to 0·55 per cent carbon content the pads have been satisfactorily hardened to Rockwell C.56 to 58 in approximately 13 sec. For this purpose a plate type coil measuring 1½ in by 1½ in by 14 s.w.g. and having a 1/32 in centre hole has been found satisfactory. Details of this coil are shown in Fig. 4 which also indicates a simple method of handling rockers on a production basis. The exact depth of the hardened zone will naturally be a matter for individual requirements, but it can be varied by adjusting the amount of power applied to the pads. In the case referred to above, the hardened zone varied from 0·16 in to 0·21 in.

Valve stems

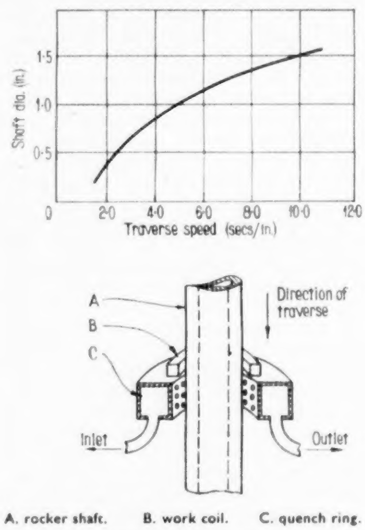
As mentioned in the foregoing section, the rocker pads operate directly on the top of the valve stems and if wear is to be avoided it is desirable that the ends of the valve stems should also be hardened. The depth of hardness will vary from one valve manufacturer to another, but this again can be controlled when using induction heating.

Although fairly elaborate jigging arrangements have been made in some instances, the arrangement shown in Fig. 5 has proved to be entirely satisfactory. In this arrangement, the work coil consists of a three turn solenoid, the centre turn of which is slightly smaller than the two outer turns. The valves are placed in the "V" blocks which position them in the correct relationship to the centre line of the work coil and which also act as stops to ensure that the correct length of valve stem enters the coil. The operator places two valves in the "V" blocks and judging the temperature by eye removes them and immediately places them in a quench bath arranged at a convenient position underneath the heating coil. He then picks up two new valves and repeats the process.

In the above arrangement the power can be left on continuously as the heating time for valves having a stem diameter of 0·3125 in is in the region of 3 sec with a 6 kW equipment. This time is increased to 5 sec if a 2½ kW heater is used.

Valve forging

An interesting application of induction heating is to be found in the production of the forging for automobile valves. The normal procedure in the matter of valves of this type is to take a length of bar metal, resistance heat one end and upset it by applying pressure to the heat zone and this has the effect of shaping the bar metal into a roughly hemispherical form. Whilst still hot, the upset bar is passed to a

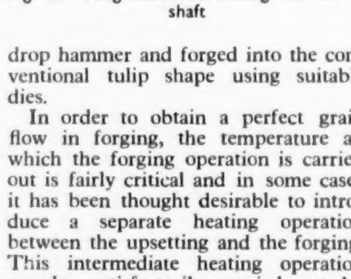


A. rocker shaft. B. work coil. C. quench ring.

Fig. 3. Progressive hardening on rocker shaft

drop hammer and forged into the conventional tulip shape using suitable dies.

In order to obtain a perfect grain flow in forging, the temperature at which the forging operation is carried out is fairly critical and in some cases it has been thought desirable to introduce a separate heating operation between the upsetting and the forging. This intermediate heating operation can be satisfactorily carried out by induction heating, the method being to cool the bars after the upsetting oper-



A. plate coil. B. brazed-on cooling tube.
C. stop bar. D. spring loaded locating plunger.
E. rocker. F. mounting shaft.

Fig. 4. Set-up for hardening rocker pads

ation and then introduce them into the coil of a high frequency generator for a given period of time so that the temperature of the valves when they are placed in the forging die is in the region of 1200 deg C. Tests carried out on valves in the upset stage indicate that the desired temperature can be obtained on a valve with a $\frac{5}{16}$ in stem upset to 1 in diameter in approximately 3-5 sec using a 15 kW induction heater.

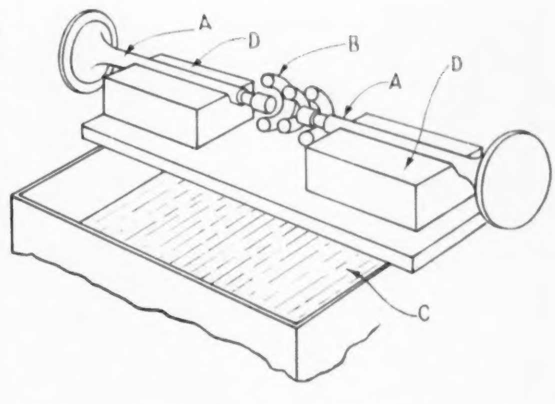
Fig. 6 illustrates a simple method of handling valves on a production basis. It will be seen that the handling equipment consists of a dished turntable in which a series of slots are cut around the outer edge. The upset valves are hand loaded into these slots as indicated and the serrations on the base plate cause the valve to rotate. When the top table is in position a suitable type of tunnel coil is arranged adjacent to the table so that the valves are at the correct temperature when they leave the coil and at this point the base is cut away and the valves fall off into a shoot leading to the forging press.

Brazing pipe unions

In the construction of automobile engines, it is usual to encounter a number of unions which have to be brazed on to oil or fuel pipes. The brazing of such unions can be carried out successfully and economically by induction heaters of low power output. Fig. 7 shows a typical arrangement for this operation.

With a $2\frac{1}{2}$ kW equipment the brazing of a brass union to a $\frac{1}{4}$ in copper tube can be accomplished in 17 sec. If a 6 kW heater is used this time can be reduced to 13 sec and in both cases the time required for brazing can be further reduced by utilizing an external boosting condenser in conjunction with the induction heater. It is usual for the solder in the form of wire to be made into a ring of suitable dimensions and placed inside the union prior to the insertion of the pipe and with a work coil similar in construction to the one shown, it is possible to obtain an even flow of solder over the entire joint faces.

Although many concerns using induction heating for this type of application on a production basis do not utilize any handling fixtures, it is possible to employ simple locating jigs to ensure that the union and pipe are held in correct relationship to the work coil (see Fig. 8). In some cases, for example in the assembly of oil gallery tubes for lubricating the main bearings of an engine, it is not possible to use a simple solenoid type coil owing to the fact that the projections on the various unions prevent the work from being withdrawn from the coil. In such cases it is possible to utilize hinged coils which allow this type of work to be successfully carried out.



A. valves. B. three-turn solenoid coil. C. quench tank. D. vee blocks.
Fig. 5. Arrangement for hardening valves

Sump drain plugs

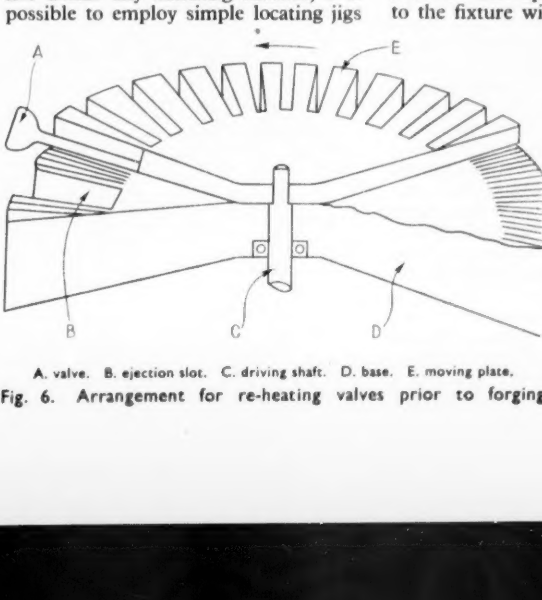
Modern motor car engines produced on mass production lines generally employ a steel pressing for the engine oil sump.

In order to enable the oil to be drained from the sump with the engine in position a screwed plug is generally provided at the lowest point of the sump. To secure this plug in a thin steel pressing, a screwed boss must be provided. This boss usually takes the form of a screwed steel or brass ring which is brazed on to the pressing. Excellent results have been obtained using a $2\frac{1}{2}$ kW heater to carry out the brazing operation referred to above.

In the case under review, however, a steel boss was brazed to the pressing using a silver solder with a 630-640 deg C melting range, and a tensile strength of 28·5 tons/in^2. The time necessary to produce a satisfactory joint on a production basis was 20 sec. A fairly elaborate brazing fixture was utilized. It is illustrated in Fig. 10 from which it will be seen that the fixture consists of a two station turntable which is provided with locating blocks for the sumps.

The sequence of operations is as follows: The operator loads a sump on to the fixture with the drain plug boss and silver solder ring in position, he applies flux and operates a push button which rotates the table through 180 deg. This brings the drain plug immediately underneath the work coil and the whole table is then automatically elevated by means of a pneumatic cylinder, thus bringing the parts to be heated into the correct relationship with the heating coil.

Whilst the work is being heated, the operator mounts a new sump on to the second locating fixture and at the end of the heating cycle, the turntable is automatically lowered and is indexed by the operator by means of a press button. The sequence is then repeated.



A. valve. B. ejection slot. C. driving shaft. D. base. E. moving plate.
Fig. 6. Arrangement for re-heating valves prior to forging

Suspension arms

Many independent suspension units utilize some form of wish-bone between the chassis and the wheel and in recent years it has become common practice for these wish-bones to be made from steel pressings. At the extremities of the wish-bones it is usual to provide bearings on which the wish-bones can rotate and it has been found possible to braze the bearing carrying cross tubes into the main pressings by means of high frequency induction heating.

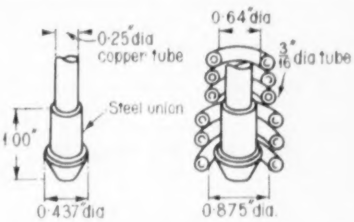


Fig. 7. Arrangement for brazing pipe unions

If reference is made to Fig. 9, some indication can be gained of the method by which the induction process is applied to applications of this type. Internal work coils are used which enable two solder joints to be made simultaneously. In Fig. 10 these two joints are marked (XX) and (YY) respectively. The handling gear is so designed that it is possible for the operator to reverse the complete assembly in order to complete all joints and in the example under review the complete brazing time for the 8 joints was 60 sec with a heater having an output of 6 kW.

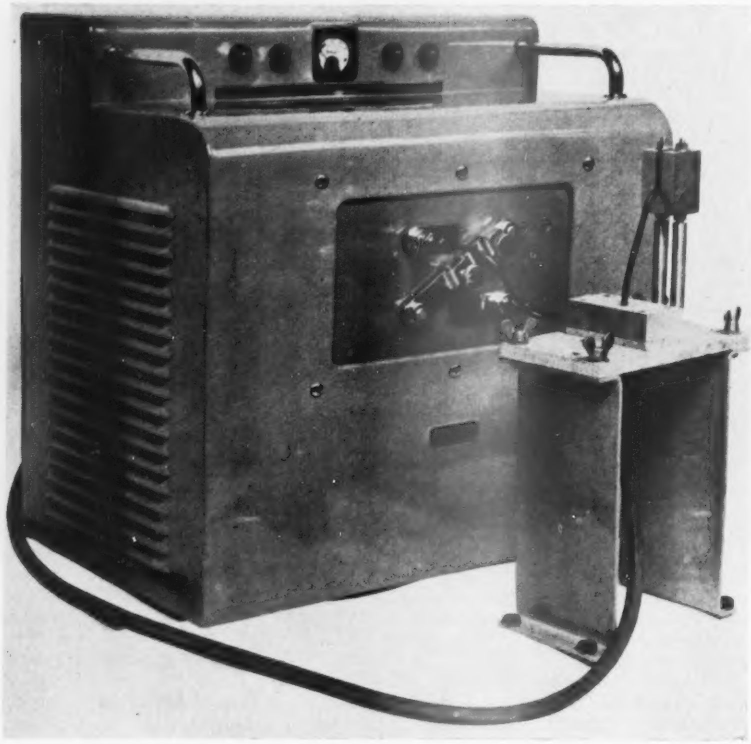


Fig. 8. Delapena induction heating equipment with simple hand-loaded fixture for brazing automobile oil pipes

To deal with this application on a production basis a two station fixture was used and the work was loaded by hand. As in the previous example, the work is raised and lowered in relation to the coil by means of pneumatic mechanism. The assemblies are passed to the operator of the brazing fixture with the bushes already pressed into position so that he has ample time to press the solder rings round the tubes, apply flux and mount the assembly on to the fixture whilst the heating is being carried out.

Steering tube assemblies

In manufacture of heavy motor vehicles, the steering column tubes are frequently fabricated from steel tubes and castings, the tubes being brazed into position. The brazing operation referred to above can be successfully carried out using induction heating.

The usual method consists of inserting several rings of silver solder wire into the tube, applying flux liberally and then fitting the tube into the casting. The assembly is placed inside a heating coil and the solder flows over the joint faces by capillary action. It is of interest to note that the production time required, using a simple multi-turn solenoid coil in conjunction with a 15 kW heater, is in the region of $2\frac{1}{2}$ min.

Windscreen wiper shafts

The driving shafts for windscreen wiper motors are typical of the numerous small steel components used in the automobile accessories and in common with many other such parts, these have to be hardened. For one application the shafts were turned from $\frac{5}{16}$ in diameter carbon steel suitable for direct water hardening. At one end of the shaft teeth were cut to take the drive from the motor. It was necessary to harden the whole shaft with the exception of the teeth to a hardness of Rockwell C.64-65. To carry out this operation on a production basis, a $2\frac{1}{2}$ kW induction heater of the valve oscillator type was used and the time required to heat the shafts to quench temperature was 5 sec.

A hopper type handling gear was used. The shafts are fed into the hopper and a feed slide carries one shaft at a time into the delivery chute which is so designed that the shafts are tipped into correct position before arriving at a platform at the bottom of the chute. The platform is connected to a solenoid and when it is withdrawn, it allows the shaft to fall from the chute into the heating coil.

Underneath the heating coil a second platform is arranged which is also controlled by solenoids and when withdrawn this in its turn allows the heated shaft to fall into a quench ring and then through another sliding platform into a cooling bath to complete the quenching operation. In this particular case the high frequency power was kept in operation continuously, the feeding of the shafts and the operation of the various platforms being simultaneously to maintain a steady production based on 5 sec required to heat the shafts.

Fuel injectors

In construction of fuel injectors used on compression ignition engines a hardened and ground steel face on the injector body abuts the face nozzle proper. This face on the injector body has to be hardened and it has been common practice to carburize locally and harden it by separate operation. In this method, however, the screwed thread adjacent to the face is hardened at the same time.

Induction heating was first introduced into this sphere with a view to annealing the screwed threads after they had become hardened as described above. In this manner it was possible to utilize the localized heating effects of induction heating and still leave the joint face hard. For this operation a simple solenoid coil was used to surround the threaded portion of the body. Further investigations indicated that with a pancake type of coil the face only could be hardened, but eventually the pancake coil was replaced by a plate coil.

Using a heater of 15 kW output it was found possible to harden the joint faces of the injector bodies to Rockwell C.58-62 in 7-8 sec and to achieve a case depth of 0·027 in. It was found that by increasing the heating time to 8 sec the depth of case increased to 0·036 in. In both cases the depth of the hardness layer was even over the entire surface area.

If heating were applied for only 5-6 sec, the case depth achieved was not

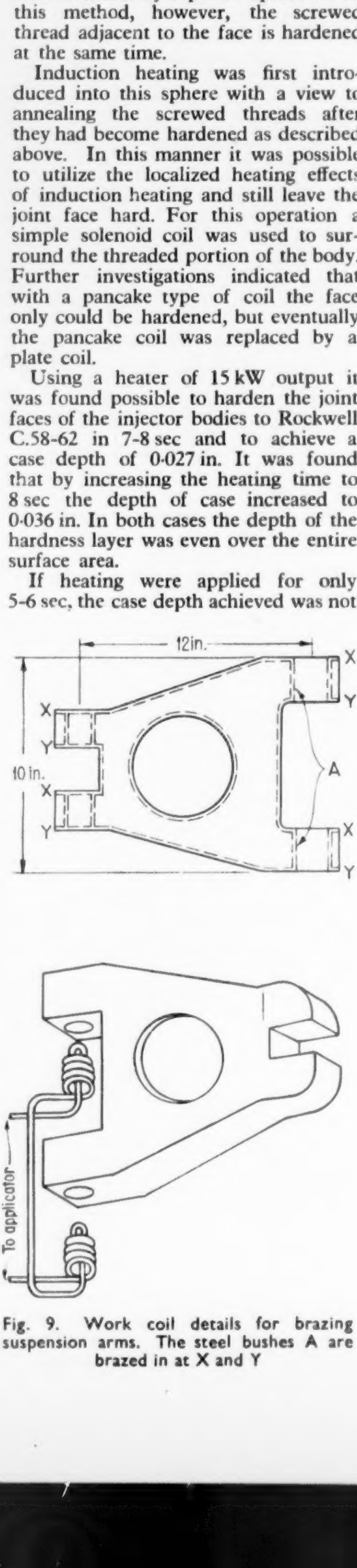


Fig. 9. Work coil details for brazing suspension arms. The steel bushes A are brazed in at X and Y

sufficient and those heated for 10 sec developed cracks during the quenching operation. Quenching was carried out by means of a quench pipe so designed as to direct water through the hole in the split coil. For handling the injectors on a production basis a simple turntable type of fixture was used in which a locating device held one injector body in the correct relationship to the work coil. During the heating cycle other injectors are hand loaded on to the turntable by the operator.

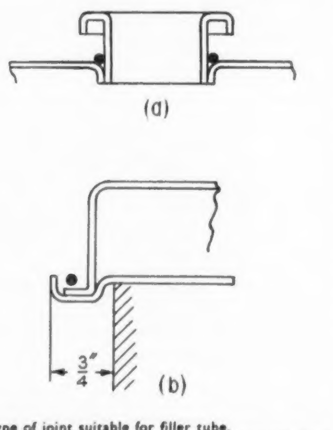
Radiator construction

A particularly successful application of high frequency heating is to be found in the soldering of automobile radiators and some indication of the suitability of the equipment for this purpose will be gained when it is realized that the majority of manufacturers of these components now utilize induction heating.

When considering the possible application of this form of heating to radiator construction, it is generally necessary in the first place to examine the design of the radiator in order to determine whether the joints are suitable for the use of high frequency heating. In some cases it may be necessary to introduce modifications to the form of joints, but it has been found that such an action is well justified in view of the very substantial savings both in production and material costs which are obtainable.

In general it should be realized that when utilizing high frequency induction heating the solder is applied in the form of rings of solder wire placed adjacent to the joint faces prior to the application of heat. Because of this it is always desirable to provide a ledge or pocket to support the solder ring and typical examples of such joints are shown in Fig. 11.

A further point to be borne in mind when examining the design of radiators with a view to the utilization of induction heating is the method to be employed to bring the work coil adjacent to the joint; for example, some designs incorporate outlet tubes from the top or bottom tanks which are curved to such an extent that it is not possible to bring the work coil in the correct relationship to the joint faces.



a. type of joint suitable for filler tube.
b. details of joint for soldering top and bottom tanks for radiator core.

Fig. 11. Radiator joints

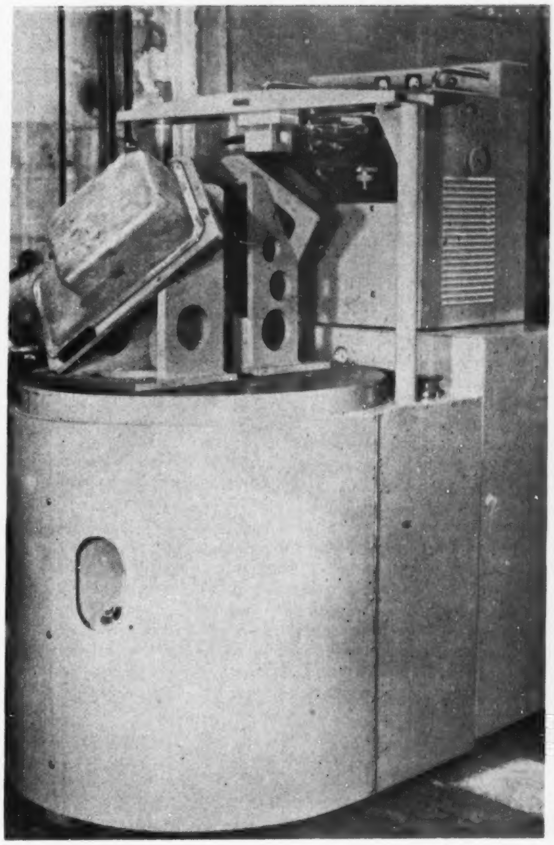


Fig. 10. Delapena equipment with semi-automatic hand-loaded brazing fixture for engine sumps

It must be stressed that the process is not suitable for soldering the radiator tubes into the top plate of the core block, but all other joints such as the securing of the top and bottom tanks to the core, the soldering of inlet and outlet tubes and the soldering of filler caps, etc., are generally ideally suited to the process.

One of the major advantages of the process, apart from very appreciable savings in production time, is that by utilizing preformed solder rings very considerable savings in solder consumption are made possible and in practice these savings may amount to as much as 50-60 per cent compared with other conventional types of soldering. With the extremely high price of solder, it will be appreciated that the above economies are very significant.

It is not possible to give exact details of the work coils used, as these will vary with each individual design of radiator, but it is of interest to note that in the case of the large joints around the top and bottom tanks a simple fixture carrying a hinged work coil can be employed. It will be seen from the illustrations given in Fig. 11 that a general recommendation is given that the distance between the tank joint and the radiator core should be $\frac{3}{4}$ in and the reason for this is that in some cases where this dimension is smaller, there is a danger of the solder in the core assembly becoming melted under the action of the work coil. If it is not practicable to maintain these recommended dimensions, a possible solution lies in the use of a higher melting point solder on the core as compared with the solder used for the tank joint.

With regard to production times, these again will vary very considerably from one type of design to another, but it is significant that in the case of a certain type of radiator the time required for soldering the complete radiator assembly was reduced from 1 hr 20 min to 3 min using a 6 kW equipment. For larger radiators it may be advisable to use induction heating equipments with somewhat larger power outputs.

Shock absorber bodies

An example may be given of the methods for brazing automobile shock absorber bodies of the telescopic type. These bodies are manufactured from steel tube into which are fitted end caps to which are secured the fastening bosses. It has been found possible to braze the end caps into the tubes and also to braze the securing boss to the end cap by means of high frequency induction heating.

Fig. 12 illustrates diagrammatically the method employed. The various parts of the assembly are prepared in the following manner: The inside of the tube is treated with flux and a single ring of 18 s.w.g. silver solder is pushed into the mouth of the tube. Flux is then applied to the end cap which is pushed into the tube until it is in its correct position. The steel boss is next treated with flux and is placed in the groove provided in the end cap, a ring of 18 s.w.g. silver solder having first been located around the boss. The form of the silver solder ring is important. If the ring is not a good fit around the

groove, it is not possible to make a good joint.

The assembly prepared as described above is then placed in a suitable fixture made in such a manner that the work to be brazed is held in the correct relationship to a work coil consisting of 4 turns of $\frac{3}{16}$ in copper tube, the top turn of which encompasses the retaining boss of the shock absorber. With this arrangement it is possible to braze the 2 joints simultaneously and using a 6 kW heater a satisfactory result can be obtained in 25 sec. It is, of course, possible to reduce this time if a more powerful generator is utilized.

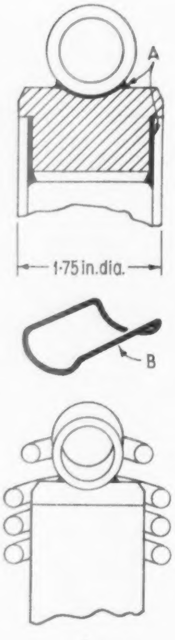
Shackle pins

The shackle pins used on the suspension units of automobiles have been successfully case-hardened using the induction heating method and using a 15 kW heater in conjunction with an output transformer and a single turn work coil, a 0·050 in case has been successfully produced on pins 0·625 in in diameter. The pins were hardened progressively by passing through the work coil and were quenched immediately afterwards by a suitable quench ring assembly. In order to carry out this hardening operation, a fixture similar to that shown diagrammatically in Fig. 13 was used and the following brief description may be of interest.

The shackle pins are fed from a hopper which automatically positions the pins for feeding into a safety box which is provided with a simple device which ensures that should a pin be fed into the box the wrong way round, the heater is automatically switched off and the mechanism of the handling gear stopped. The method of operation of this device can clearly be seen from Fig. 13. After leaving the safety box, the shackle pins pass through a guide which positions them in the correct relationship with the work coil. Heating of the pins occurs and quenching is provided by means of a suitably designed quench ring situated immediately underneath the work coil. After being quenched, the pins pass through a second guide block which, like the first, is provided with a ceramic liner.

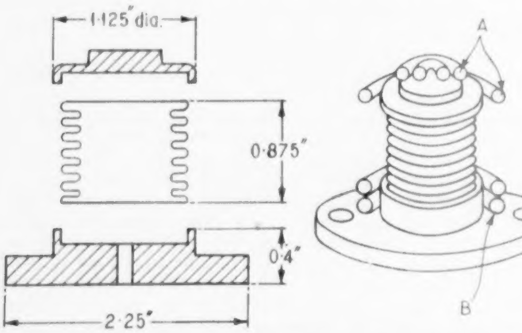
Underneath the second guide block there is a large cam and the end of the shackle pins rest on the cam face as shown in the illustration. As the cam rotates, the shackle pins fall at a predetermined rate which determines the depth of case obtained. Suitable switching arrangements for the high frequency power are provided to coincide with the passing of the threaded and tapered portions of the pins passing the work coil.

In order to prevent the danger of jamming, a clamping block is provided which ensures that only one pin at a time can be fed to the ejecting face of the cam which delivers the treated pin to a suitable delivery chute. The clamping block is operated by means of a secondary cam secured to the driving shaft of the main cam. The arrangement is such that two spring loaded jaws prevent the shackle pin in the clamping block from being released until the top face of the cam has passed underneath the shackle pin in the lower guide block. This type of handling fixture can, of course, be used for other cylindrical components which have to be case hardened.



A. brazed joints B. form of solder ring for bush C. work coil details

Fig. 12. Brazed shock absorber bodies



A. three-turn coil, two turns form pancake and the third embraces the cover plate B. two-turn solenoid coil

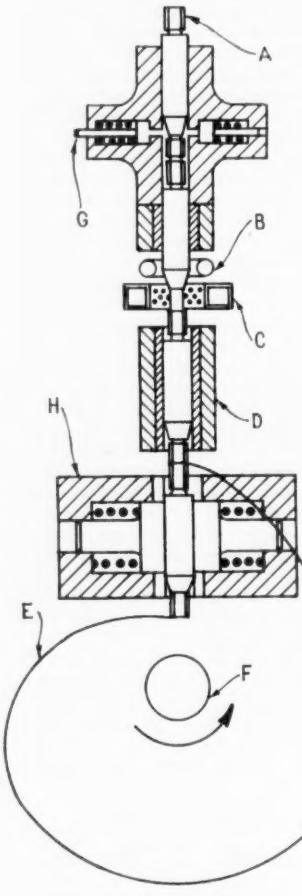
Fig. 14. Soldering thermostat bellows assembly

Brazing oil seals

In the construction of automobiles oil seals of the bellows type are sometimes employed. These seals consist of a copper, brass or stainless steel bellows element to the ends of which are brazed brass or mild steel rings. Until fairly recently it was common practice to carry out this brazing operation by means of a gas/air flame, but more recently the operation has been successfully accomplished using low powered induction heaters.

The table given on the following page indicates the comparative times for brazing different types of oil seals using gas in one case and a $2\frac{1}{2}$ kW induction heater in the other case.

An examination of the figures given in the table reveals that very substantial savings in production time can be effected by the use of high frequency induction equipment and in addition to this, it has been found possible in practice to utilize female labour instead of skilled male labour with corresponding reductions in wage costs.



A. shackle pin fed from hopper
B. work coil
C. quench ring
D. guide block with ceramic liner
E. cam
F. camshaft
G. plunger to operate generator switch
H. clamping block

Fig. 13. Set-up for progressive hardening on shackle pins

Soldering thermostat bodies

A common type of thermostat which is used on automobile and similar cooling systems consists of a flexible metal bellows. Volumetric changes in the air

inside the bellows brought about by changes in the temperature of the cooling water in the system cause the bellows to expand or contract and this movement is used to control by-pass valves, shutters or other means of controlling the temperature of the cooling system.

For incorporation in thermostat assemblies of the type described, it is necessary to close the bellows assembly at each end and this is done by means of soldering on end caps which may be either steel or brass. Generally the bellows are made from brass having a thickness of 0·005 in to 0·010 in although the end caps may be of heavier gauge material and the soldering of the end caps provides an excellent application for low powered induction heaters.

It will be appreciated that individual designs of thermostat assemblies will vary considerably in detail but the majority of the soldering work encountered is well within the capabilities of induction heating. As in all soldering work, cleanliness is of primary importance and should be borne in mind during the preparation of the work. A typical example of this type of assembly is shown in Fig. 14. In this case the two soldering operations were accomplished with a $2\frac{1}{2}$ kW heater. The joints cannot be soldered simultaneously owing to the gravity effect on the solder on the uppermost joint.

Item	Detail	Time	
		Gas	RF
1	Brass Bellows $4\frac{3}{4}$" OD × $4\frac{1}{2}$" ID	20 min	57 sec
2	Stainless steel bellows $1\frac{1}{8}$" OD × $\frac{3}{4}$" ID	16 min	16 sec
3	Brass Bellows $1\frac{1}{8}$" OD × $\frac{3}{4}$" ID	15 min	17 sec

In the case under review two solder rings were located in the end cap which had previously been fluxed with Baker's Fluid. It may be that some manufacturers will insist on the use of non-corrosive flux, but this need not constitute a difficulty. The solder rings were prepared from normal 60:40 solder wire of 18 s.w.g. (0·048 in) section.

For both joints a simple locating fixture was employed. This consisted of a suitable locating block to hold the work in its correct relationship to the work coil. The blocks are made from any suitable heat resisting material. Means are provided to subject the joint to a low pressure to ensure that the joint faces are in contact when the solder becomes fluid. The times of 9 and 14 sec for the two joints are representative for this type of work although naturally different assemblies may fall outside these figures. Any work involving similar soldering work may be considered to be an excellent application of high frequency induction heating.

In the foregoing sections of this article it has only been possible to deal with a few representative cases of typical applications of high frequency induction heating in the field of automobile production. It is hoped, however, that the cases which have been quoted will enable a general appreciation to be obtained of the scope of the smaller type of equipment.

There is no doubt that there are many components in a modern motor car which can satisfactorily be treated by this process and in view of the very considerable savings both in time and material which can be effected, production engineers should keep this process in mind when planning the production of their products.

BODY CORROSION

IN a paper entitled "Corrosion of Automobile Bodies" by F. L. LaQue, *S.A.E. Preprint*, March 3-5, 1953, corrosion in automobile bodies is considered with reference to the corrosive medium, the metal that is corroded, and the design features which help to combat corrosion. Automobile bodies are adversely affected by the effects of the atmosphere and of materials thrown up from the roads. The corrosive attack is influenced by atmospheric humidity and amount of rainfall, and by particles of dust, ashes and soot. On iron and steel, the adverse effects of the atmosphere can vary enormously in severity with the locality; it is generally more severe in heavily industrialized areas and coastal areas. The *p*H of rain water can be as low as 4, and the dilute acid solutions present in both rain and dew may cause high rates of corrosion. Heavy rainfall and high humidity seem to be less injurious than fog and heavy dew. For treating icy roads, salt is preferable to cinders, especially if inhibitors are added to the salt. Where salt is used, valuable precautionary measures are the frequent washing of cars to remove adherent salt which is hygroscopic, and ensuring that cars not in use are kept in a dry place.

Steels for car bodies are not primarily selected for their corrosion resistance. A low copper content, of under 0·05 per cent, associated with a relatively high sulphur content, gives particularly poor resistance. Appropriate amounts of alloying elements such as phosphorus, chromium and nickel can considerably improve the resistance to atmospheric corrosion. A phosphate treatment of the metal before painting gives further improvement. Most corrosion of automobile bodies begins at the inner surfaces to which adequate protective coatings should be applied.

Body design should be such as to ensure that water cannot collect in pools inside. Water may get in the body as a result of rain entering around windows, moisture condensing at night from air drawn into confined spaces by day, or liquid thrown up from wet roads into cavities in the body. Arrangements should be made so that all surfaces may drain readily and dry quickly. Collected water is readily removed by providing outlet holes where necessary; surfaces may also be inclined so that water will drain down to the holes. The quick drying of steel surfaces may not be so easy; in motor coaches, controlled heating and ventilation are the solution. For the drying of under-bodies of cars, an arrangement is suggested for conveying air heated by the engine to these critical parts. *M.I.R.A. Abstract No.* 6275.

ARDOLOY TOOLS

TO assist production engineers in keeping abreast of modern developments in cutting carbides, demonstrations with Ardoloy tools will be carried out at the Edgwick Works, Coventry, of Alfred Herbert, Ltd., from the 14th September for a period of at least two weeks. The aim of the exhibition is to demonstrate the durability and cutting qualities of Ardoloy tools on modern machines. All kinds of Ardoloy tools will be in use, and the machines will be set-up for actual production work.

Standard lathe tools will be used for single and multi-cutting operations from light finishing cuts to extremely heavy cuts absorbing some 50 h.p. Polytip tools will also be shown in action to demonstrate the remarkable increase in output between regrinds that can be obtained in comparison with lathe tools. In some applications the increase has been truly remarkable.

Chipstream turning and parting-off tools will be set up on machines for high production bar work. The Chipstream boxtool is well known for its ability to produce at high speed a constant diameter with a good finish. Microbore tools will also be shown on precision boring, turning, facing and chamfering operations.

CRANKSHAFT FORGING

A Highly Mechanized American Plant

A REMARKABLY mechanized forge section has recently been brought into operation by The Chrysler Corporation for the production of press-forged crankshafts. In comparison with the methods previously used, the new plant has made it possible to produce forgings two to three times as quickly. The outstanding feature of the plant is the very wide use made of automatic mechanical handling devices.

An elaborate system of electrical, mechanical and hydraulic devices, time controlled or actuated by various trips or photoelectric cells, maintains the flow of materials. Nearly eleven miles of electric wiring is used in the system for the various elements that handle and move materials; and for operating the forging furnace, the main and supplementary furnaces, the heat-treatment furnaces and the numerous blowers.

Three items of equipment are of particular interest. First is a giant, 6,000 ton, high-speed mechanical forging press, which is capable of more than 35 strokes a minute. The second is a rotary, doughnut type, billet heating furnace, which is loaded and unloaded automatically. Thirdly, there are the automatic hardening and tempering furnaces.

A new building, 462 ft long and 150 ft wide houses the new press forge shop. It is of a type of steel and masonry construction that provides the maximum natural light. In addition, there is an advanced type of forced ventilation that not only ventilates the whole building but also provides individual, adjustable forced air outlets at each working station. For moving equipment about the plant there is a 50-ton overhead crane arranged to traverse the whole length.

Fig. 1. Automatic withdrawal mechanism on billet heating furnace

Briefly, the continuous press forging sequence is: Steel billets, stored in a yard outside the plant, are lifted by a magnetic crane into a rack in the plant wall. These billets, 4-4½ in square, are then pushed by a hydraulically actuated pusher-rod into a pre-heating furnace. This furnace is used principally during cold weather to heat the billets to as much as 400 deg F to avoid any danger that cracks will be caused during the subsequent shearing process.

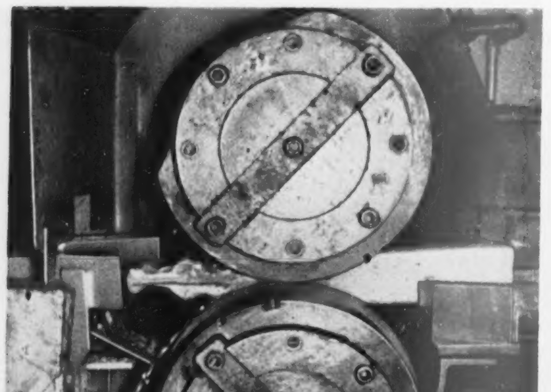


Fig. 2. Billet passing through reducing rolls

When the billet leaves the pre-heating furnace, it falls on to a conveyor which moves it into a No. 15 shear, which is automatically set in operation when the billet contacts a trip. The shear cuts the billets into pieces that range in weight from 106 to 125 lb, dependent upon the type of crankshaft into which they are to be forged.

From the shear the billet is transferred by mechanical conveyor to a rotary furnace. When the billet reaches the appropriate position, a hydraulically-operated, jaw-type mechanism picks up the billet and moves it into the furnace. The loader carries the billet into the furnace and then lowers it gently to the furnace floor. This eliminates the wear on the furnace floor that cannot be avoided when loading is effected by pushing the billet into the furnace. The loading mechanism can be pre-set to handle billets at rates up to 300 per hour. This furnace is gas started and oil-fired, with burners playing from both sides towards the billets. The temperature is automatically maintained at 2250-2300 deg F.

Each loading action of the rotary furnace is followed by a movement of the furnace floor sufficient to provide space for the next load. This process is continuous and the heating cycle takes approximately one hour. The unloading operation, which is carried out adjacent to the loading station, is performed by a mechanism similar to that used for loading, but operating in reverse. The unloading device is shown in Fig. 1. The complete sequence of feeding, loading, movement of the furnace floor and unloading is electrically synchron-

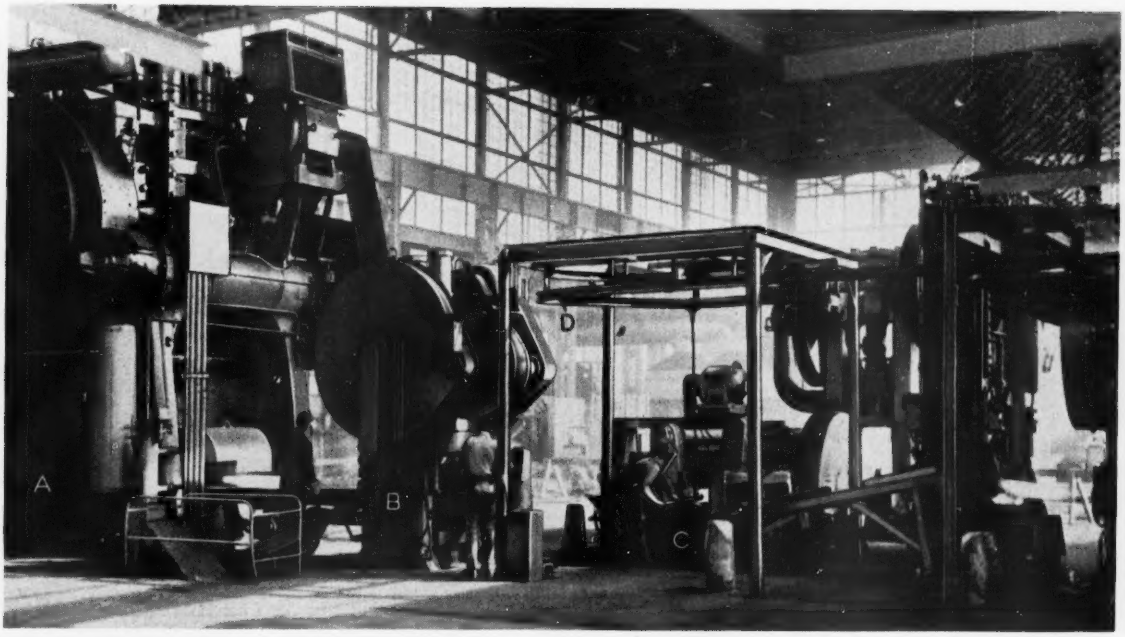


Fig. 3. Press forging section

ized and hydraulically and mechanically actuated.

As the heated billet is taken from the furnace it is placed on a gravity roller conveyor down which it slides to a turntable rotating through 90 deg. From the turntable the billet is mechanically conveyed through a four-jet water de-scaler that operates at between 1500 and 2000 lb/in^2 water pressure. From the de-scaler the billet is conveyed to a No 10 reducer which, by rolling, distributes the material of the billet into the positions needed for forging. A billet passing through the rolls is shown in Fig. 2. The rolls are automatically set in operation when the billet contacts a trip.

For the actual press forging operation, which is next in the sequence, it is necessary to turn the billet end-over-end through 180 deg to bring it into position for the press. A turn-over device is installed immediately behind the reducer rolls. When the billet leaves the rolls it causes a photo-electric cell to actuate an air-driven piston that turns the billet over to the desired position. The work is then conveyed mechanically to the press section shown in Fig. 3. This illustration shows the forging press A, the trimmer B, the upsetter C and a monorail conveyor D.

Two-impression dies are used for press forging, and only one strike is required on each die. From the forging press the shaft is conveyed to an adjacent trimming press and excess metal is trimmed off. After it has been struck, the forging is pneumatically ejected from the trimmer. The trimmed-off material is placed on a slide, see Fig. 4, leading to a hole in the shop floor. The scrap falls on to an automatic conveyor beneath the floor to be conveyed to the scrap pile.

Three more machines complete the actual press forging plant. First there is a four-inch upsetter for forging the flange to a degree of accuracy that will greatly minimize the amount of machining that will be required subsequently. The next machine in the sequence is an hydraulically-operated twister, see Fig. 5, which is used only for V.8 crankshafts. These shafts have to be twisted to bring the crankpin bearings into proper alignment. Six-cylinder engine crankshafts are in

Fig. 4. Removing surplus metal at the trimming press

Fig. 5. Twisting press for V.8 crankshafts

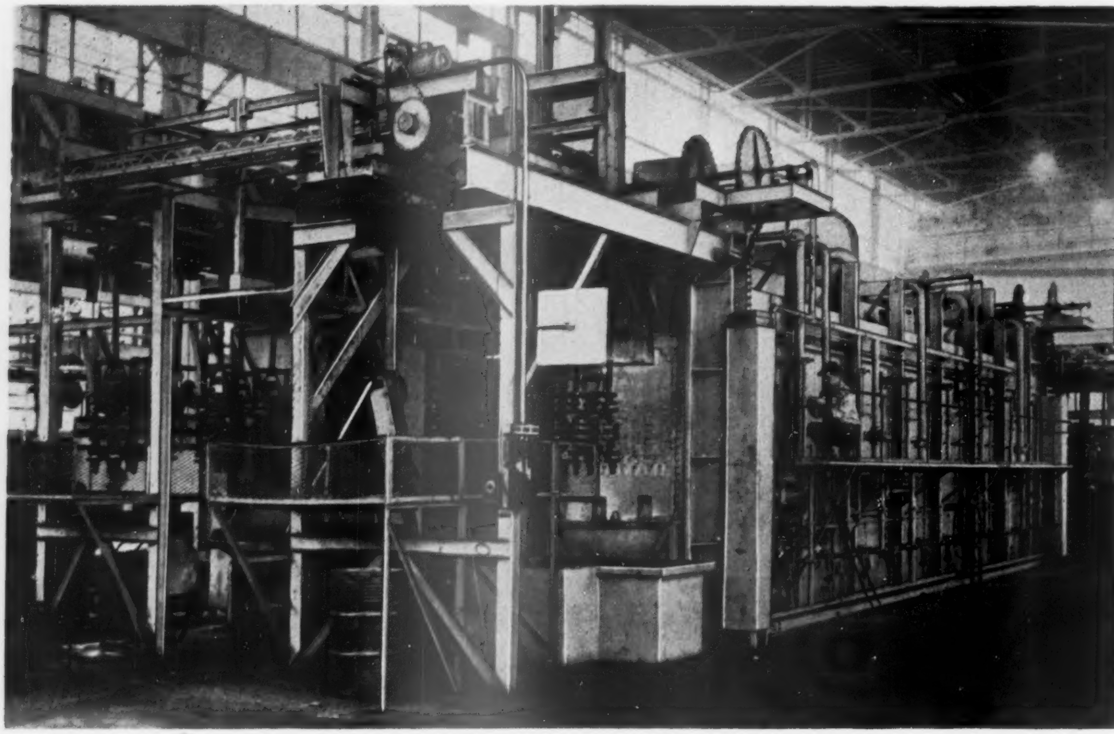


Fig. 6. The hardening and tempering furnaces

proper alignment when they leave the forging press. Finally, the forging goes to a re-strike press for straightening. The work is conveyed from one press to another in this sequence suspended from a trolley on an overhead monorail.

After the straightening process the shaft is placed on a rack from which it is picked up by an intermittently operated overhead conveyor on which it moves through an air cooling zone to the hardening furnace. As the shafts approach the hardening furnace, they are manually transferred to carriers by means of a pneumatic hoist. The carriers operate from a monorail handling system. Each one accommodates four crankshaft forgings suspended vertically. An air hoist elevator moves the loaded carriers either to a storage bank or to a supply station where they are put into position to move into the furnace in groups of two. The hardening and tempering furnaces are shown in Fig. 6.

Carriers enter the hardening furnace at intervals of just over three minutes. The furnace temperature is 1550 deg F, and the work passes through the furnace in 100 minutes. On emerging from the furnace the carriers are automatically lowered to immerse the forgings in a tank of water held at 90 deg F. To ensure uniform hardness the quenching water is agitated. The quench takes 70 seconds. At the end of this period an elevator raises the carriers from the tank and back to the monorail for transfer to a gas-fired tempering furnace. Once again, carriers in pairs enter the tempering furnace at intervals of just over three minutes. Passage through the tempering furnace is carried out in 120 minutes. Tempering is effected at 1125 deg F. As the work leaves the tempering furnace it is lowered into a tank of water to be cooled, is raised and returned to a monorail for transfer to a shot blasting cabinet for a thorough cleaning.

After the shot blasting the forgings are transferred by air hoist to a conveyor on which they pass to a centring machine. The centring operation is set up and carried out to constant dynamic balance checks. A subsequent alignment procedure straightens the forging to static balance on live centres. The shaft is then checked dimensionally and inspected for hardness and possible defects before it is shipped to the machining section.

NEW TYPE SPECTROGRAPH

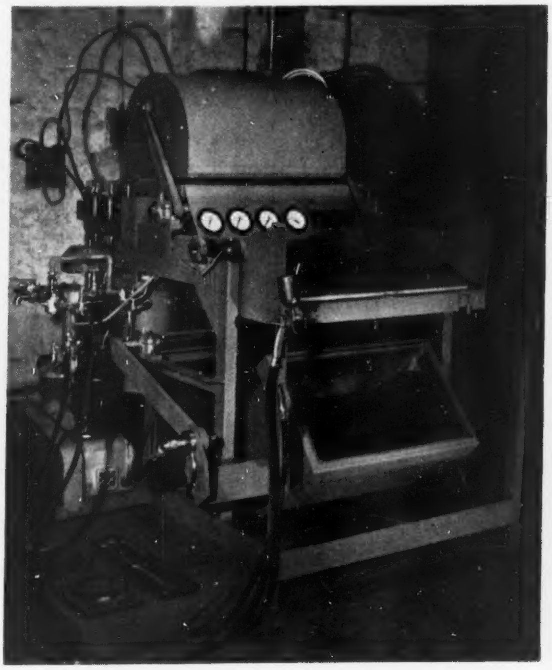
A NEW type of spectrograph is mentioned in *The Engineers' Digest* of June 1953. Since the accuracy of this spectrograph is greater than that of earlier types, it should prove a useful instrument both in research and in industrial practice. It is claimed that because of the combination of a prism and an echelle, the instrument makes possible quick routine analyses of proportions of less than one part in a million. Hitherto the trend has been to make spectrographs larger and larger to obtain maximum utilization of prisms and diffraction gratings. Space limitation, however, makes it difficult further to increase the size of spectrographs and now the echelle offers an alternative solution to the problem.

As used in the new instrument, an echelle is an inch thick, optically flat glass plate, 3 in by 5 in, in which grooves have been ground accurately and spaced to give a distribution of 200 to the inch. Like a prism or a diffraction grating, it disperses light into a spectrum so that the composition of the light-emitting source can be studied in detail. The dispersive power of an echelle, however, can be ten or more times that of a diffraction grating, and its resolving power or ability to show the separation of spectral lines is doubled or trebled. In the new instrument, the echelle is used in conjunction with a prism. The prism spreads out the spectrum horizontally on the spectrographic plate, while the echelle spreads it out vertically. (2048)

AUTOMATIC SHELL MOULDING

An Interesting New Machine

BECAUSE of its many advantages for a wide range of applications, the shell moulding process is arousing keen interest throughout the engineering industry. A recent and important development for use in conjunction with the process is the Autoclino automatic shell moulding machine shown in the accompanying illustrations. This machine was designed by A. Wansborough Jones, M.I.Chem.E., M.I.Mech.E., in association with Shaw Processes Ltd. It is manufactured by Clino Foundry Supplies Ltd., 25, Clyde Vale, Dartmouth Road, Forest Hill, London, S.E.23. It is designed to be suitable for both quantity production and for batch production in a jobbing foundry. In each case the moulds can be produced rapidly and economically.



Autoclino shell moulding machine with covers removed

A pattern area 28 in × 16 in and pattern depths up to 10 in can be accommodated. The machine cycle, which is initiated by push button control, is completely automatic and non-repeating. Compressed air is employed to supply power for all the several machine movements. It can be taken from the shop air-line at a pressure of 100 lb/in^2 or from a built-in compressor. An air receiver is built into the machine to serve as a reservoir to even out any fluctuations that may occur in the supply pressure. An air filter and a lubricator protect the system and ensure smooth valve operation.

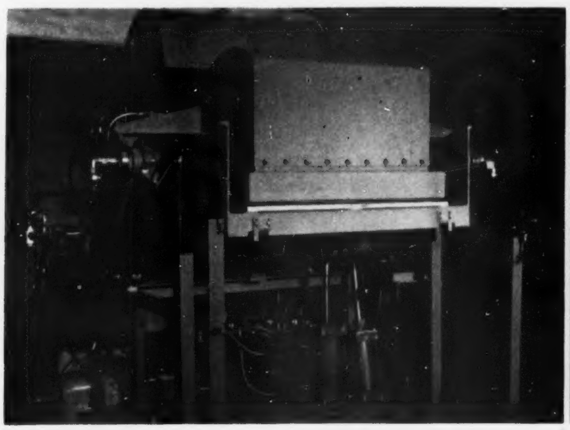
The pattern plate is mounted on a framework that is carried in substantial trunnion bearings. A gear quadrant is fitted to one trunnion member to engage with a rack which is directly connected to an air cylinder so that movement of the piston is transmitted to the rack and the pattern frame is caused to turn in its bearings. The pattern plate incorporates two spring-loaded ejector springs. A pyrometer is fitted in contact with the underside of the pattern plate. The pattern temperature is recorded on a dial on the machine.

A dump box is mounted beneath the pattern plate. It has an opening 28 in × 14 in and will take slightly more than 150 lb of sand/resin mixture. In the "at rest" position the box is tilted forward at an angle of 45 deg to facilitate loading with fresh sand/resin mixture and for visual inspection of the contents. Where contact is made with the hot pattern plate a silicone rubber gasket is fitted in a recess in the upper edge of the box.

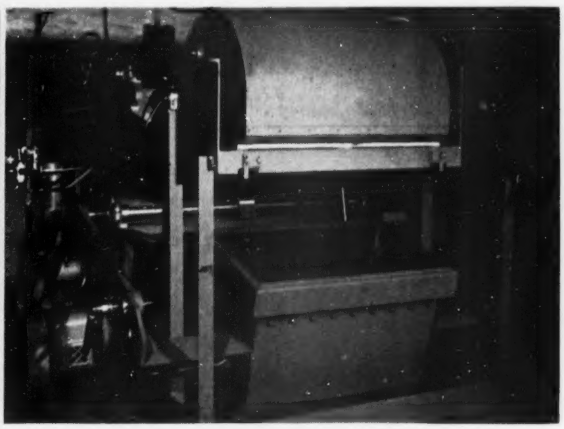
Infra-red heating elements are used for heating during the investment cycle and for curing the mould. They are mounted in a hood which, at the correct instant is autocatically pivoted forward to come to rest over the pattern. The duration of these heating periods can be accurately controlled to predetermined settings.

The operation of the machine is simple. After a preliminary run to determine the correct length for each of the various time cycles, the timing devices are set to reproduce the master cycle. Consequently the whole operation is correctly timed and will be accurately repeated indefinitely.

A master time switch controls the operation to give a fixed and unvarying duration for each motion. There are also four independent time switches which can be hand set to any given period in the range. Each of these takes over from the master switch during its own part of the operation cycle. The timing devices actuate solenoid operated



Dump box and patterns in the investment position



Infra red heating hood in curing position

air valves which control the air supply to the cylinders operating the machine.

The general sequence of operation is:—

(1) The pattern plate, which is mounted on two dowelled studs and secured by two hexagon nuts, is loaded.

(2) The heating hood swings over the pattern plate and applies infra-red heat for the predetermined period.

(3) The heating hood swings back and the pattern plate is reversed.

(4) The dump box rises to meet the pattern plate.

(5) The dump box and the pattern plate are turned completely over to allow the sand/resin mixture to invest the pattern. After a set period, this movement is reversed to allow any surplus mixture to fall back into the box, which is then lowered away from the plate. The pattern and its investment is then turned over to the original position and the heater hood swings over the invested pattern for the curing cycle. After a set time the hood returns to its stand.

(6) The dump box is lifted up to the ejector plate which then engages the ejector pins to raise the shell mould about 2 in and holds it clear of the plate for the pre-set time to allow the operator to remove the shell. The dump box is then lowered and the machine is ready for the initiation of another cycle.

The time for a complete cycle varies with different pattern plates according to the thickness of mould required. Naturally, a thick mould requires a longer curing time than a thin one, but an average cycle time may be taken as from 2½ to 3 minutes. In such a cycle the actual operational time would not be more than 30 seconds, the remainder of the cycle being taken up by the investment and curing times.

ALUMINIUM ALLOYS

Notes on Heat-treatment Equipment

ALTHOUGH all aluminium alloys do not gain in strength or hardness by heat-treatment there are many that owe their industrial importance to the mechanical properties that can be developed by the correct heat-treatment. The heat-treatable alloys are divided into two main groups: (a) those that harden spontaneously after initial high temperature treatment (known as *solution heat-treatment*), and (b) those that can be hardened still further by a second heat-treatment at a lower temperature (called *precipitation heat-treatment* or *artificial ageing*). Those in category (a) are known as single heat-treatment (natural ageing) alloys and those in category (b) as double heat-treatment (artificial ageing) alloys.

Generally, reference to the "heat-treatment" of aluminium alloys implies solution treatment with or without artificial ageing, but annealing is another thermal treatment that is in common use. It is employed for softening and improving the ductility of both work hardened and heat-treated alloys.

Solution heat-treatment

The temperature range for the solution treatment of a commercial alloy depends upon the alloy composition and it is important that the ranges recommended by the manufacturer of the material should be followed. If too low a temperature is employed, incomplete solution will result and maximum mechanical properties will not be developed, while too high a temperature will produce melting of certain constituents, which may cause cracking of the metal and serious impairment of its mechanical properties. The permissible temperature range of solution heat-treatment for a given alloy is usually not more than 10 deg C.

In solution heat-treatment it is the general practice to heat to slightly above the theoretical point of complete solution, but great care must be taken not to heat too high above this temperature.

Because strict control is essential for efficient solution treatment, special equipment is necessary. The exact requirements depend upon the nature of the work that is to be treated. Two main types of furnaces are used: air furnaces in which heated air is circulated round the load; and salt baths, containing molten salts in which the load is immersed. Both types may be heated by electricity, gas, oil or solid fuel.

Air furnaces

Air furnaces are the more generally used for heat-treating aluminium. Whatever the heating medium, the furnace should be of a design that ensures rapid, efficient and uniform transfer of heat to all parts of the load. Therefore forced air circulation produced by an arrangement of fans and baffles is usually necessary. No matter what the source, the heat provided must be sufficient to bring the load rapidly up to the temperature of the furnace, which should be maintained within 5 deg C of the approved temperature.

Electrically heated furnaces are usually preferred because of ease and accuracy of temperature control, their greater uniformity of temperature and the purity of the atmosphere in contact with the load. Gas or oil-fired furnaces are generally of more intricate design since they must be of the muffle or heat-exchanger type in which the products of combustion do not enter the heating chamber where they might cause blistering of the load.

Air furnaces are of three main types, each specially suitable for the solution treatment of a particular commodity. A furnace for the treatment of sheet consists of a horizontal chamber, usually rectangular in cross section, with forced air circulation. A quench tank is situated in front of the furnace, and rapid withdrawal quenching mechanism is provided. This type of horizontal furnace is often used for extruded sections and sometimes for castings and forgings.

It is sometimes more convenient to carry out solution treatment of castings and forgings in a pit-type furnace. This is cylindrical and sunk into the ground. The components are loaded into suitable cages and lowered into the furnace. The quench tank is situated in a pit alongside the furnace. Vertical type furnaces are suitable for large batches of tubes and long thin extruded sections. The cylindrical heating chamber is supported vertically above ground level; under it is a track with a pit below. The load is suspended from a "spider" which is lowered on to a carriage on the track and drawn under the furnace. It is then hoisted up into the body of the furnace and a cylindrical quench tank, into which the load may be directly lowered, is hauled underneath the furnace.

Salt baths

A salt bath is essentially a container in which salt baths are maintained in a molten condition at the desired temperature. It therefore comprises a tank with some means of heating, together with instruments for indicating and recording the temperature, and, usually, apparatus for automatic control.

There are several means of heating—electric immersion heaters, external electric heaters, external gas or oil firing and radiant tube—all of which are in general use. Of these, the electric immersion heaters are generally preferred.

Generally air furnaces are more convenient than salt baths for the solution treatment of wrought aluminium alloys and are used for a greater variety of commodities. Yet salt baths have considerable advantages, the chief being the higher rate of heat transfer from the heating medium to the load. These notes are based on a booklet, *Heat Treatment of Aluminium Alloys*, issued by Northern Aluminium Company Ltd., Banbury, Oxfordshire.

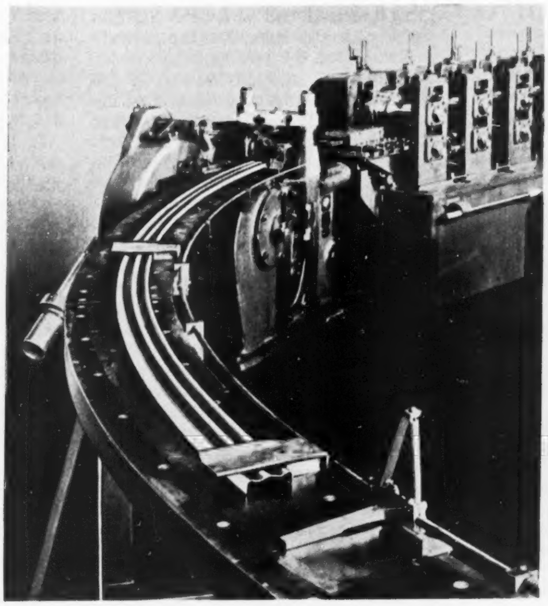
COLD ROLL FORMING

Recent American Developments

Frank Spicer

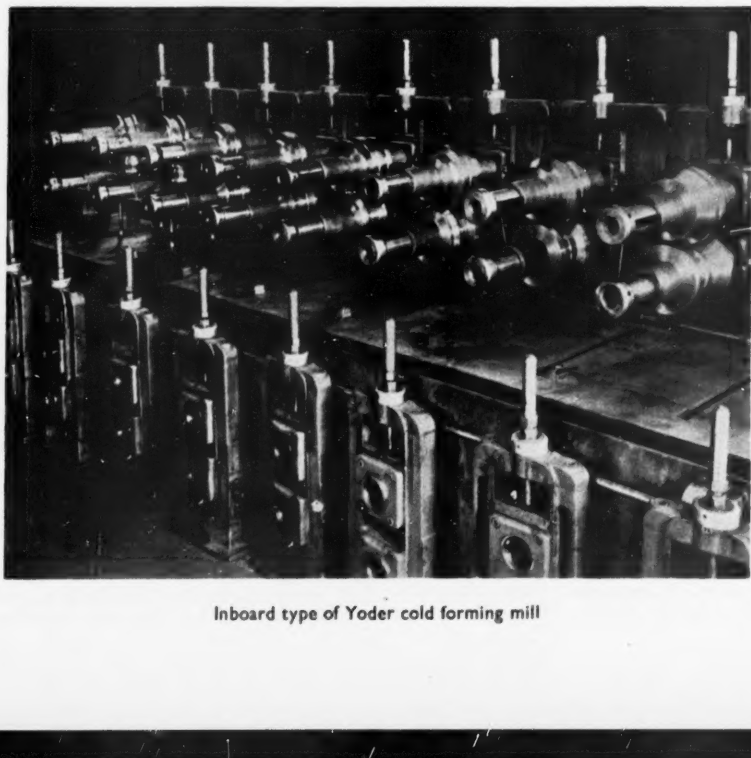
COLD roll forming is a process whereby a flat strip of metal passes through a series of rolls arranged in tandem and is progressively formed into the ultimate desired shape. The metal requires no heating or heat-treatment before, during, or after forming. The number of rolls depends upon the intricacy of the shape. The versatility of cold roll forming is well suited to high speed production, and its products are used in such widely situated industries as facing, cappings for aircraft and motor cars, furniture, structural fittings, etc. They have been a boon to the designer who wishes to achieve improved appearance and added strength.

Before the cold roll forming operation there is a sequence consisting of hot rolling, gang slitting to the desired width and recoiling for convenient feeding into cold roll forming machines at high speed. Although the primary function is to produce cold formed sections the machines are adaptable and can be fitted with attachments for levelling, trimming, spot or seam welding, cutting to length, notching, perforating, embossing, curving, coiling and ring forming. When it is realized that a machine operating at 100 feet per minute can produce 30,000 ft a day with only an operator and a labourer, it is obvious that cold roll forming is designed for low cost production. The rolls can be changed in about 3 or 4 hours and this delay is quickly neutralized by the production speeds. Hot and cold rolled carbon steels, bronze, aluminium, brass and copper are all susceptible of cold forming.



Yoder mill for cold forming curved sections

Among the foremost cold forming machines in the world are those made by the Yoder Company of America. The machines are of two types, inboard and outboard; the latter machine has its spindles supported at one side and overhung; the inboard type has the spindles supported at both ends. It is with this type that the author is familiar and will therefore discuss.



Inboard type of Yoder cold forming mill

All roll stand assemblies are standardized and are interchangeable, and the top spindles are driven through a toggle gearing arrangement which keeps the gears in full mesh over a wide range of spindle adjustments, reducing wear and tear to a minimum. The gears required for driving the main rolls can be either of equal or unequal gear ratio. When sections having deep profiles are being formed, the upper rolls have a larger pitch or driving diameter than the lower rolls, with a corresponding decrease in the revolutions per minute, in order that the peripheral speeds of both upper and lower rolls shall be the same.

It is advisable to use a lubricant for all roll forming operations, this not only lessens the wear on the rolls but also helps to prevent roll pick-up which marks the sections. One method of lubrication is to use two felt rolls constantly fed with a light oil, one continually covering the top and one the bottom as the strip passes through.

Usually the machine is fitted with a variable speed motor, although it is generally preferred to operate the machine at a constant speed, since too much variation leads to complications in setting-up, particularly where a flying shear cut-off is in operation. The size of the machine is dependent upon the type of section to be formed and the roll stands can vary from 2 to 16 sets. The width of the strip is likewise controlled

since it is obvious that a machine will have a minimum and a maximum width that can be accommodated.

Cold forming

Many cold roll formed shapes may easily be curved or formed into complete rings or spirals and then cut to length. Curving is accomplished in a die with a curved run-out table. For ring-forming purposes a number of ingenious arrangements have to be fabricated on the job; among such shapes are head lamp rings, cycle rims, etc.

It is possible to insert strips of wood, fabric, rubber, wire or tube in a roll formed channel or box shape, or to combine two pieces of metal, one laid over the other, forming them into one piece, the edges of the lighter strip being folded in around the edges of the other to hold them together. This, of course, opens up possibilities of using a thinner section of an expensive rustless steel and backing it up with a cheaper, but strong material. As an example, very thin material of 0·004 in to 0·006 in thickness of stainless steel may be rolled face to face with a strip of heavier gauge low cost steel to form almost any shape.

Printing and embossing rolls may also be incorporated. If the strip is to be embossed over its entire width, it is done on the flat strip and usually in the first pair of rolls. If only a part of the strip width has to be embossed over a fairly flat area it may also be done on the first roll pass, but if the pattern is to be deeply embossed the embossing rolls should be placed last, to avoid flattening by the forming pressure.

To avoid stopping the machine when one coil of strip has been used up and another coil has to be inserted, a simple looping mechanism, a butt welder and other equipment can be incorporated so that the lead end of a new coil can be welded to the tail end of the preceding one. While the two ends are being joined there is enough footage in the loop to permit continual feeding of the machine, and by the time the jointing operation is complete the loop has been taken up and fed into the machine. The ends can also be manually welded, but this, of course, slows down the production rate.

Even with a 16 roll machine, only the number strictly necessary to produce any given section are used. As the strip passes through the machine, each pair of rolls produces a partial change in the cross section, consequently the number of rolls necessary will depend upon the intricacy of the section and the kind of material to be formed. The complete equipment for a machine will not only include the rolls, but also the spacers for positioning the rolls on the spindles, straightening device, idler rolls and shears for trimming the strip as it enters the machine.

The bar type guides are designed to keep the strip in proper alignment as it is fed into the first pair of forming rolls, and to keep the strip free from any vertical or horizontal deflection. In all cases whether using coil or cut-to-length stock, the feed table is fitted with these guide bars. Idler rolls are so called because, unlike the main rolls, they are not power driven. They do, however, take part in the forming process, supplementing that done by the main rolls. These idler rolls differ from the main rolls in that they operate on a vertical axis and are adjustable mainly on the horizontal plane—just the opposite of the main rolls.

A straightener is always attached to the exit end of the machine, close to the last roll pass. It is designed to prevent the finished section from turning and twisting. However this straightening device should never be used to rectify inaccuracies in roll set-up and alignment. Shearing of the strip into lengths is done by a flying shear. As the section leaves the machine it passes through a die of the same profile, mounted on a movable table. The shape progresses along the run-out table until the front end comes into contact with a trigger, which brings the cutter down on the section in the die. The die table during the cutting operation moves forward with the section until it is completely severed.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Industrial Brazing

By H. R. Brooker and E. V. Beatson, B.Sc., A.M.I.E.E.

London: ILIFFE AND SONS LTD., for *Welding and Metal Fabrication*, Dorset House, Stamford Street, London, S.E.1. D8vo. 344 pp. Price 35s. 0d.

This is the first full-length, authoritative study of the brazing process as applied to metal fabrication. It covers in considerable detail all modern brazing methods. That is, torch furnace, high frequency, induction, resistance, salt bath and dip methods are all described. Full consideration is also given to the special techniques necessary for work on aluminium, stainless steels, beryllium copper, cemented carbides and vacuum tube construction. In addition, a general introduction to the processes and the equipment employed is followed by a review of brazing materials. There is also a most complete summation of the known factors that govern joint design. The problem of selecting the most appropriate brazing process for different types of work is discussed, and there are sections on testing and inspection. A selected bibliography enhances the value of this book.

This work will be of value both to those engaged in the actual industrial application of brazing processes and to those concerned with the design of brazed assemblies. More than 200 photographs and diagrams are used to illustrate various phases of the several processes, and a great amount of reference material is provided in both the text and the 32 tables.

The High-Speed Internal-Combustion Engine

By Sir Harry R. Ricardo, LL.D., F.R.S.

Glasgow: BLACKIE AND SON LTD., 17, Stanhope Street, Glasgow, C.4. Fourth Edition. 1953. 6¾ × 9¾. 420 pp. Price 40s.

The author of this book needs no introduction, for he is one of the best known and most experienced authorities on internal combustion engines. That he has a thoroughly practical outlook on the subject is reflected in almost every one of the 420 pages and 19 chapters of the work. He has aimed most successfully at bridging the gap between the scientist and the practical engineer. In this respect the book is most valuable, for the scientist is well versed in the natural laws but is generally hazy as to the capacity of the ironmongery at his disposal, whereas the practical engineer, who is often uncertain of, and even overawed by, the natural laws, is well versed in the other aspects of the problem.

Technical books often tend to be heavy reading, but this is certainly not the case with "The High-Speed Internal-Combustion Engine," which, to engineers, is undoubtedly most interesting and easy to read. Possibly the book will be attractive to many because it contains practically no mathematics, but its value both as interesting reading matter and as a work of reference certainly arises from the fact that it records the author's experience during almost a lifetime of work on the subject, and that he has aired his own views instead of confining himself to factual detail.

It will be remembered that the first edition was published in 1923, the second in 1931 and the third in 1941. However, the fourth edition is not just a revision of the earlier ones; on the contrary, the author has made an entirely fresh start and rewritten it completely. He has not, as he did in the previous editions, dealt at great length with fuel and combustion for this is a book intended for the engineer and not for the petroleum technologist. Rather has he concentrated on those aspects of the internal combustion engine that have not been given much prominence in current literature.

Particular stress has been laid on the subject of mechanical efficiency which, it is suggested, should receive more attention than is often the case in current designs. Sleeve valve engines are also dealt with in some detail, and the illustrations of these, as well as all the other illustrations in the book, are both clear and comprehensive. The chapters on mechanical design, pistons and cylinder wear are of particular interest.

ELECTRIC FUEL PUMP

A Bendix Unit Particularly Suitable for Operation under Extreme Climatic Conditions

FEW modern motor vehicles are designed to operate without a fuel pump. This is because the low lines currently adopted make it impossible to provide a gravity feed by the incorporation of the petrol tank above the level of the carburettor. Moreover, some regard a rear positioned fuel tank as being desirable since the fire risk to passengers is less than with the tank installed above the dash. Probably the most widely used type of pump is the mechanically operated unit actuated by an eccentric on the camshaft. Although pumps of this kind are satisfactory on many installations, they are not always so. In some cases, where the fuel system is subject to effects of heat radiated from the exhaust system, or where there is a large suction head between the pump and the tank, trouble may be experienced from vapour lock, particularly in extremely hot climates.

By fitting the electric pump near the tank so that the fuel in the pipeline from the tank to the carburettor is under pressure instead of suction, these difficulties are obviated. Another advantage of the electric pump is that it starts operating as soon as the ignition is switched on. This is particularly useful under very cold conditions, since the carburettor float chamber is filled before the engine is turned by the starter motor, and there is no need for hand priming.

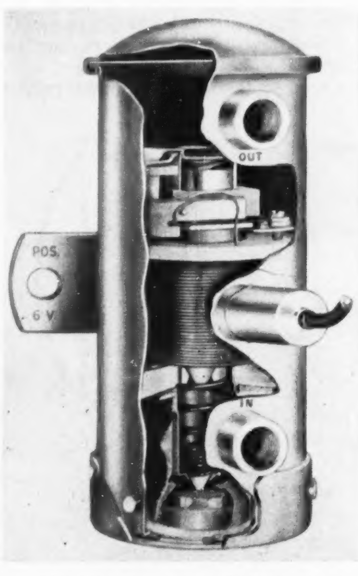
The design and development of the new Bendix pump was started in 1946 in the Eclipse Laboratory. Many designs were made and tested and improvements incorporated until the pump finally emerged in its present form. The Bendix Corporation states that the unit has now been tested under most severe operating conditions and that its reliability and long life under arduous duty have been proved beyond doubt.

In the tubular steel housing of the unit there are four sealed compartments. The lower one forms the fuel inlet chamber; immediately above that is the chamber containing the solenoid, and the electrical contacts. At the top of the unit there is a fuel outlet chamber; above it an accumulator is formed in the domed cap that closes the top of the housing. The fuel inlet and outlet chambers are connected by a brass tube which is positioned vertically in the centre of the unit.

The lower chamber is closed by a cap which is passed over the bottom end of the housing. On the sides of the housing, three pegs are projection welded. They engage in three slots in the sides of the cap. These slots are inclined in such a way that as the cap is turned clockwise, they ride up on the pegs to tighten it up against a sealing washer under the turned in lip round the lower end of the tubular housing. There is no positive means of locking the cap in this position, where it is held only by the friction between it and the sealing washer. This is a most unusual arrangement.

In the centre of the cap, a circular magnet is housed in a cage which in turn is contained in a shroud that is open at the top and welded to the cap at the base. The shroud is positioned so that fuel entering the intake valve of the plunger and pump tube assembly must pass over the magnet, which will attract any magnetic particles that may be suspended in the fuel.

The shroud also locates a cylindrical, wire filter screen that prevents the passage of all but the smallest particles of foreign matter to the carburettor. This filter consists of two screens, one of coarse mesh to give it the necessary strength and the other, of fine mesh, positioned inside the first, to act as the main filter element. Fuel enters the chamber through a boss welded on to the side, and passes from the outside of the filter to the centre. There is a similar boss on the side of the outlet chamber at the top of the unit, both bosses being tapped to receive an adaptor for a pipe union. The top of the filter assembly bears against a flanged cup which seats in the upper end of the filter screen, and in which is housed the coil spring that returns the plunger to its uppermost position.



The Bendix electric fuel pump

Between the lower end of this spring and the base of the cup is a thimble housing containing a non-return valve. The valve assembly comprises an inverted mushroom type centre piece, made of nylon, that seats in a dimpled hole in the base of the spring cup. A hole punched in the top end of the thimble housing forms a guide round the upper end of the stem, and the sides of the housing are partly cut away so that the fuel may pass freely through it. Round the stem is a small spring bearing at one end against the end of the guide, and at the other end against the head of the valve to hold it down on its seating.

The flange round the upper end of the cup containing the plunger spring, together with a thick washer of soft rubber, is pulled by three self-tapping screws in blind holes, up to the top end-plate of the lower chamber. The function of the rubber washer is to form a seal round the top of the cup which, together with the lower end of the pump tube, forms a chamber between the lower non-return valve and another in the base of the hollow plunger.

Round the periphery of the end-plate, a petrol-tight seal is made by sweating it to the housing, and at its centre is a hole in which is sweated the lower end of the brass plunger tube. Above this end-plate and round the tube is a simple solenoid winding contained in another chamber. A resistance wire is incorporated in this coil, to prevent arcing of the points. It does this by offering a lower resistance path for the collapsing flux when the points open. The coil is connected to the external wire through a fused glass and metal seal soldered in the boss provided on the outside of the case. A cap, with a grommeted hole in the centre is pressed over the boss. The grommet through which the cable passes, prevents the entry of water and other foreign matter to the external connection of the fused glass sealing member, and protects the cable from damage.

The second chamber is in two parts, the upper part containing the electrical contacts. The central tube extends up to the top of this chamber and round its upper end is a pressed brass plate that closes the top of the compartment. Sealing is effected in the same manner as before by sweating the outer periphery to the wall, and likewise by sweating the top of the tube in the central hole. Immediately

above this plate is another, the centre of which is dished so that it will spigot into the upper end of the pump tube. A hole is punched in the dished portion to allow petrol to pass up from the pump into the outlet chamber. Mounted in the top of the plunger is a buffer spring to prevent the plunger from making a noise when its motion is arrested at the top of the delivery stroke.

The hollow plunger is of mild steel, and its lower end houses a non-return valve similar to the one at the bottom of the spring cup. For several reasons, no attempt has been made to maintain close clearances between the plunger and its tube. Had a small clearance been called for, it would have made the unit unnecessarily expensive; small particles of foreign matter in the fuel would have been liable to jam the mechanism; and the friction to be overcome during operation of the plunger would have been much greater. Moreover, a small amount of wear, being a relatively large proportion of a small clearance, would have made an appreciable difference to the output, but with a large clearance, the design pressure is well maintained throughout the life of the unit. This pressure is governed by the strength of the return spring.

Tungsten contacts are employed. One is mounted on an insulated block on the base of the contact chamber and the other is on a leaf spring riveted to an arm of a U-shaped bracket. The bracket is positioned horizontally with its arms one on each side of the central tube. It is pivoted near the base of the U by two pins, passed transversely one through each arm and pressed into a block surrounding the tube.

Secured to the base of the U and extending above it is a bracket carrying a permanent magnet. When the plunger is in the uppermost position, the magnet is attracted towards it, causing the U-bracket to move on its pivot and close the contacts. This energizes the coil which draws the plunger downwards against the action of the return spring. The plunger thus moves out of the field of the permanent magnet, which is then attracted back to the case and base plate, and causes the bracket to pivot, opening the contact points.

The contact chamber is filled with helium which, being an inert gas, prevents erosion of the points so that no servicing on this part of the unit is necessary throughout the whole of its life. Moreover, because of the total enclosure of the contact mechanism, and because of the inert gas filling, there is no fire hazard.

The action of the unit is as follows. When the solenoid is energized by the closing of the electrical contacts, it pulls the plunger downwards. The non-return valve in the base of the spring cup closes and the valve in the base of the plunger opens so that fuel may pass from the spring cup to the output side of the plunger. As the plunger approaches the bottom of its stroke, the electrical contacts open, and it is then returned by the spring. During the upward stroke, the plunger valve is closed so that the fuel above it is forced into the outlet chamber, and at the same time the spring cup valve is opened so that fuel may pass from the inlet chamber into the cup below the plunger.

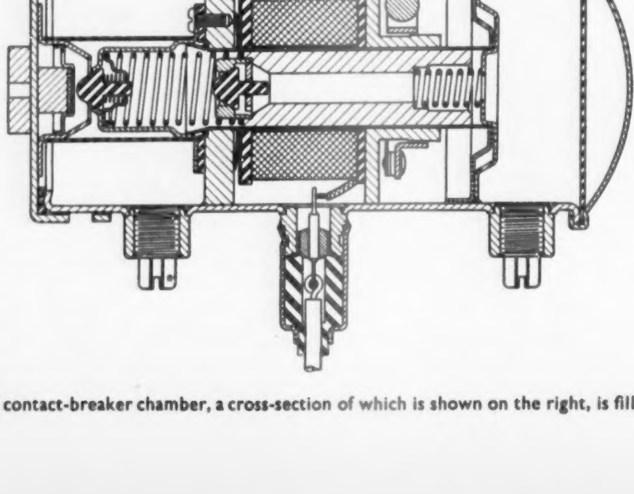
The accumulator chamber is formed between the domed cap that closes the top end of the housing and a rubber diaphragm which is clamped between the cap and the outwardly lipped upper end of the housing. Its function is to prevent unduly large fluctuations of pressure in the system, to ensure the smooth working of the pump, and to give the unit the flexibility necessary to allow for variations in the quantity of fuel required by the engine as the operating load changes. Sealing is effected by turning under the lip the periphery of the cap. From the illustration it can be seen that there is no rim moulded round the edge of the diaphragm as is usually the case with this type of component to prevent it pulling away from between the two faces that hold it.

One of the most important requirements of a pump of this kind is that there shall be no leaks either from the two chambers containing petrol or from the helium filled chamber. For this reason, air pressure is applied to the various chambers in the pump four different times during construction. The chambers are pressurized, and the unit is immersed in water so that leaks would be easily detected. In addition to this, the usual resistance and continuity checks are carried out on the electrical circuit to ensure that there is no foreign matter between the points and that there are no other defects in the system.

The weight of the complete unit is 1 lb 13 oz. Its delivery pressure may be varied from 3 lb/in^2 to 7 lb/in^2 by the use of suitable plunger springs. When the unit is operating at full capacity, the maximum power requirement is only 7 watts, so that it is not likely to add appreciably to the load on the battery. The pump is easy to install and service, since the only attention normally required is the cleaning of the filter from time to time.

The pump was first designed with automobile fuel systems in mind, but it has since been used for a wide range of other applications. It is employed in military vehicles for supplying fuel to personnel heaters. Moreover, because of its light weight, it is also used for aircraft cabin heaters. Another application for which it has been adapted is for pumping water in cooling systems serving aircraft cabins. When used for this purpose, a stainless steel plunger is provided to avoid corrosion.

Six basic models of the pump are made. These are the 6 volt, 12 volt and the 24 volt units operating with a positive earth, and units of the same voltages but with a negative earth. Moreover, there are the alternatives, already mentioned, of a standard or stainless steel plunger, and the different plunger springs for different output pressures. The pump may also be supplied with a radio suppressor or with the simple lead-in.



The contact-breaker chamber, a cross-section of which is shown on the right, is filled with helium to prevent erosion of the points

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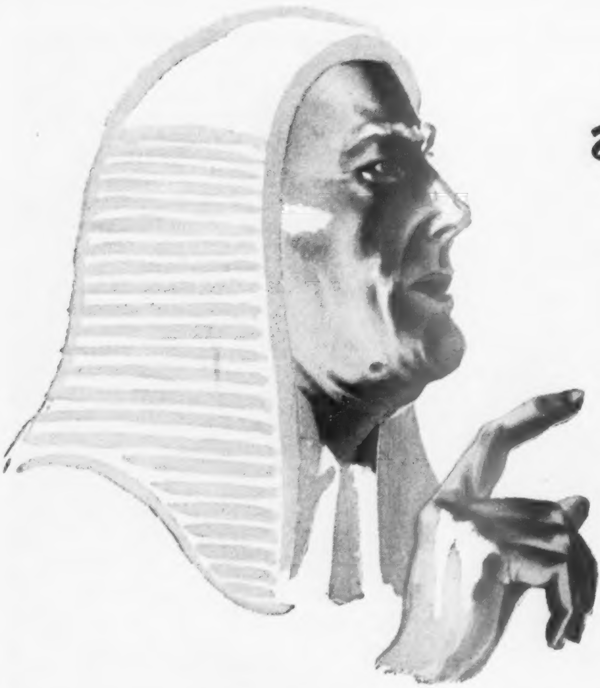

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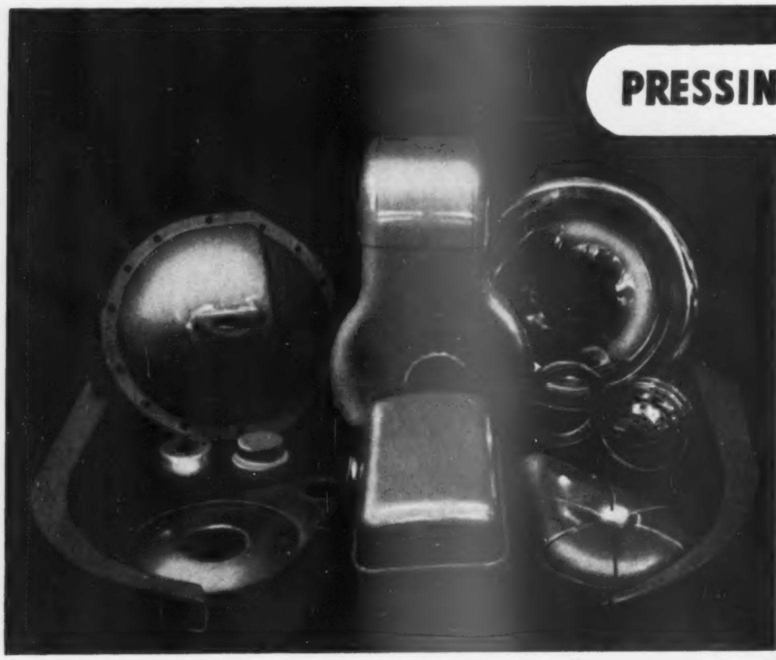

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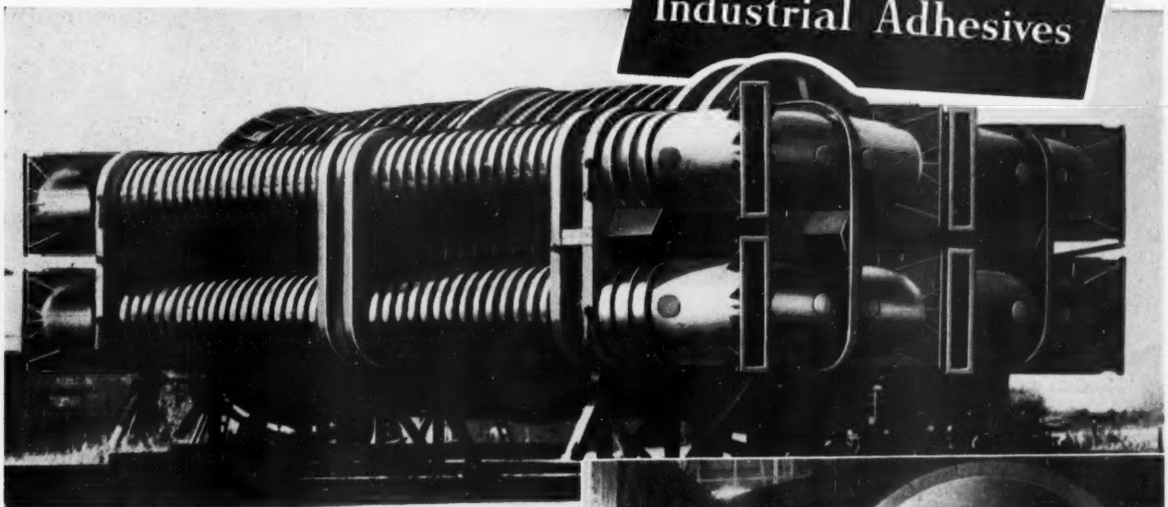

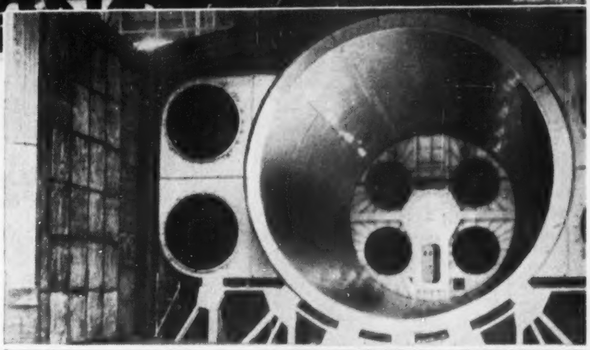

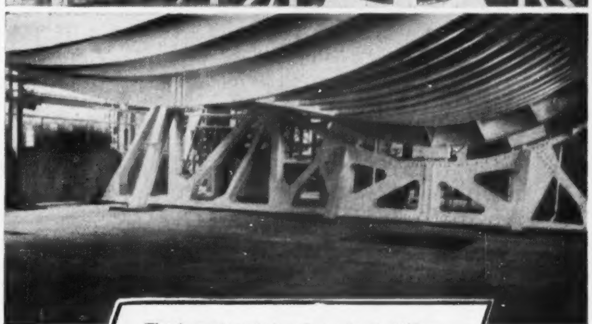

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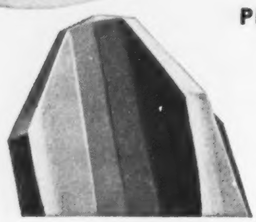

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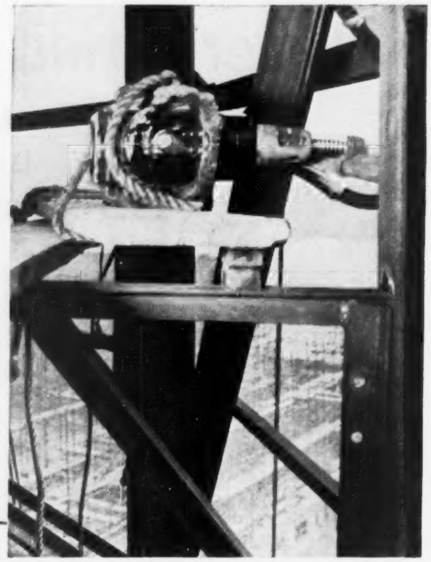

Illustration shows Newton Victor Raymax 140 kV. Industrial X-ray Unit lashed in position for radiography of welds during construction of the welded heat-storage tower for the Pimlico District Heating Scheme.
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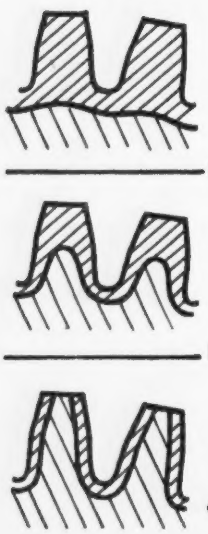

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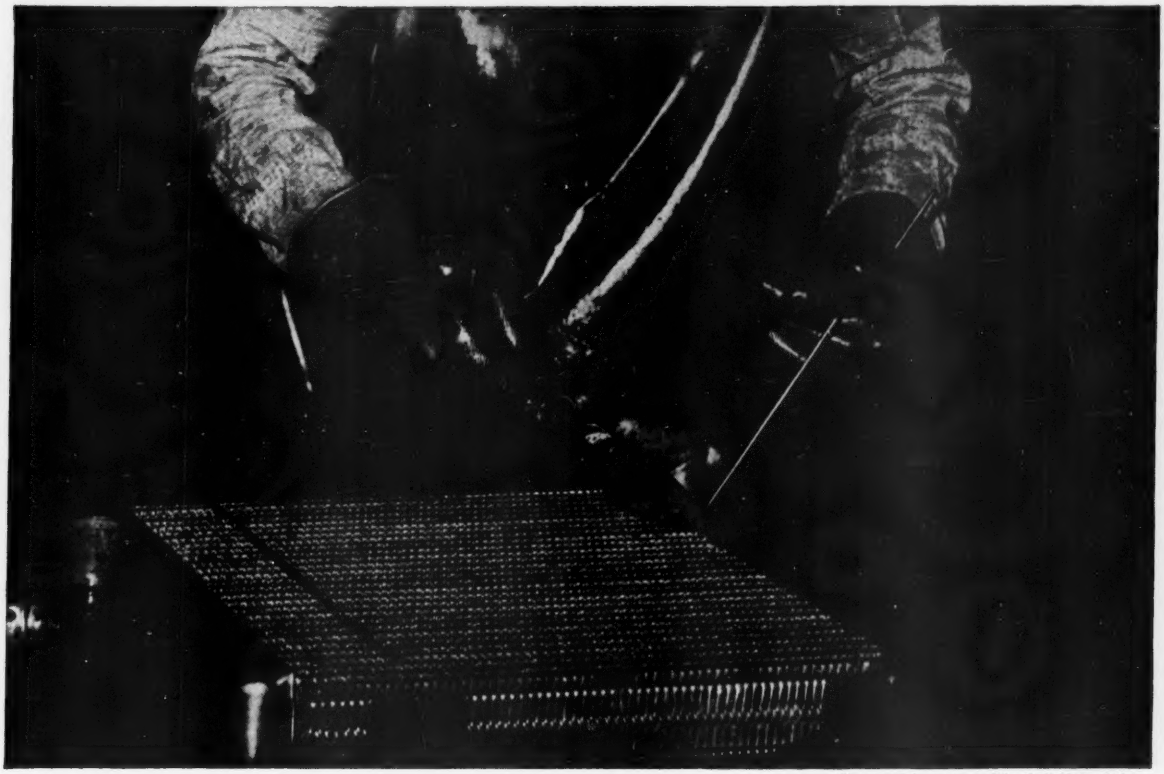

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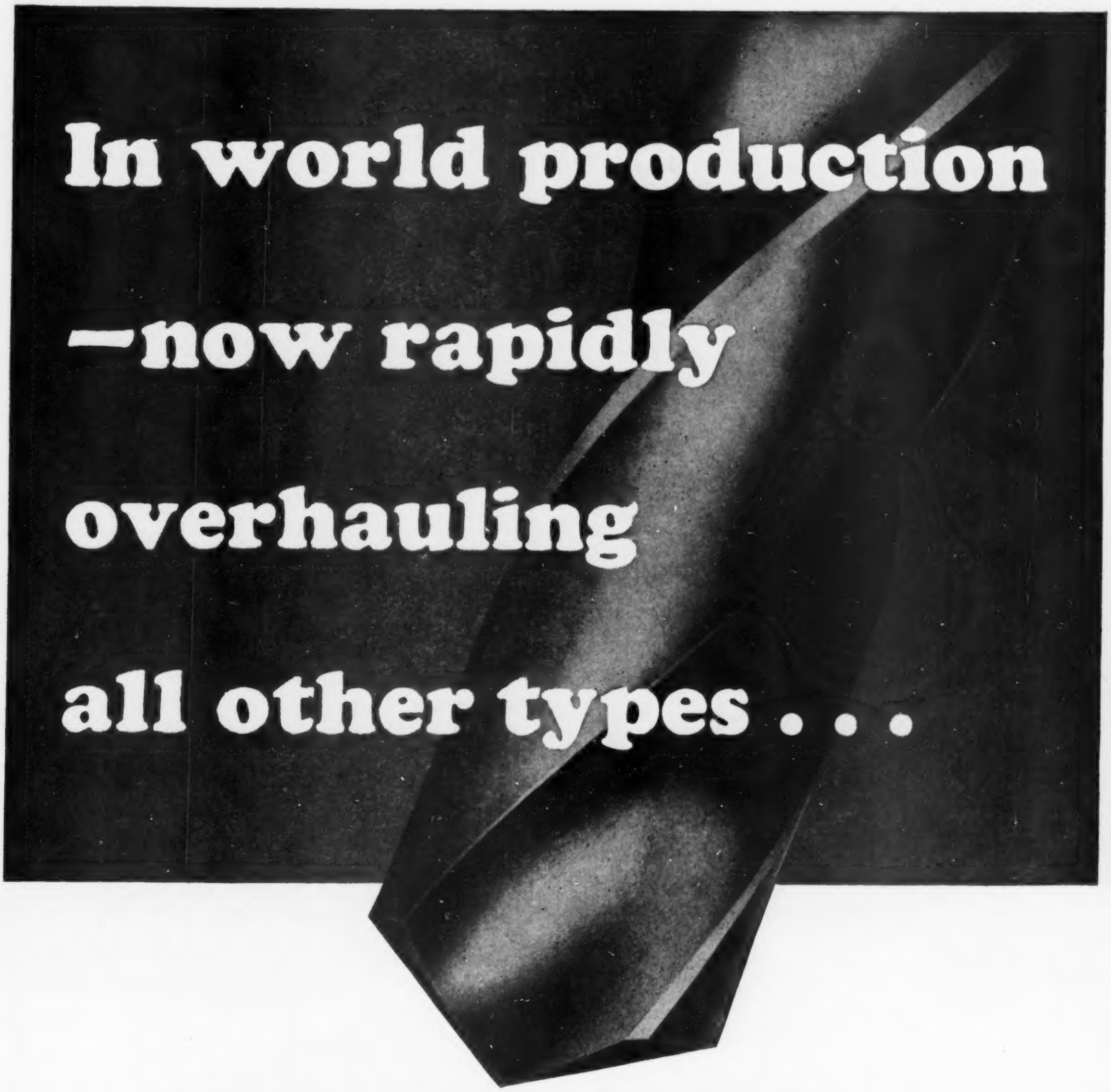
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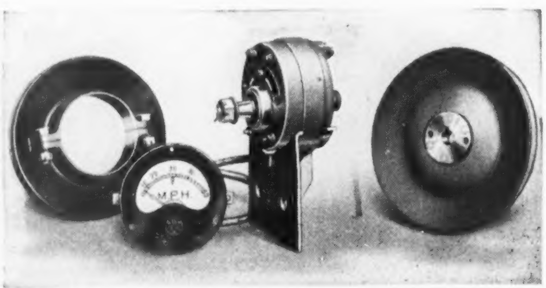

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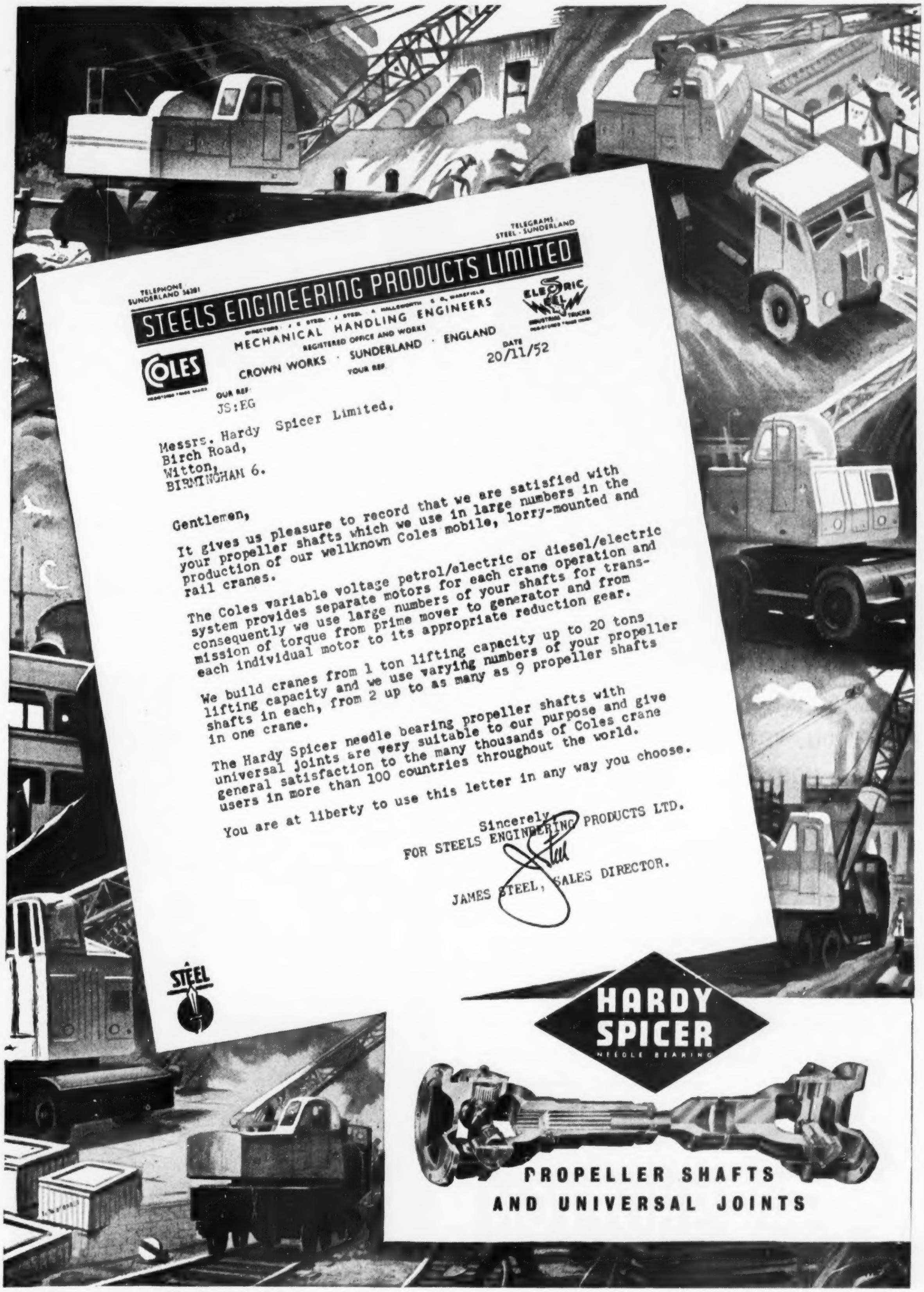

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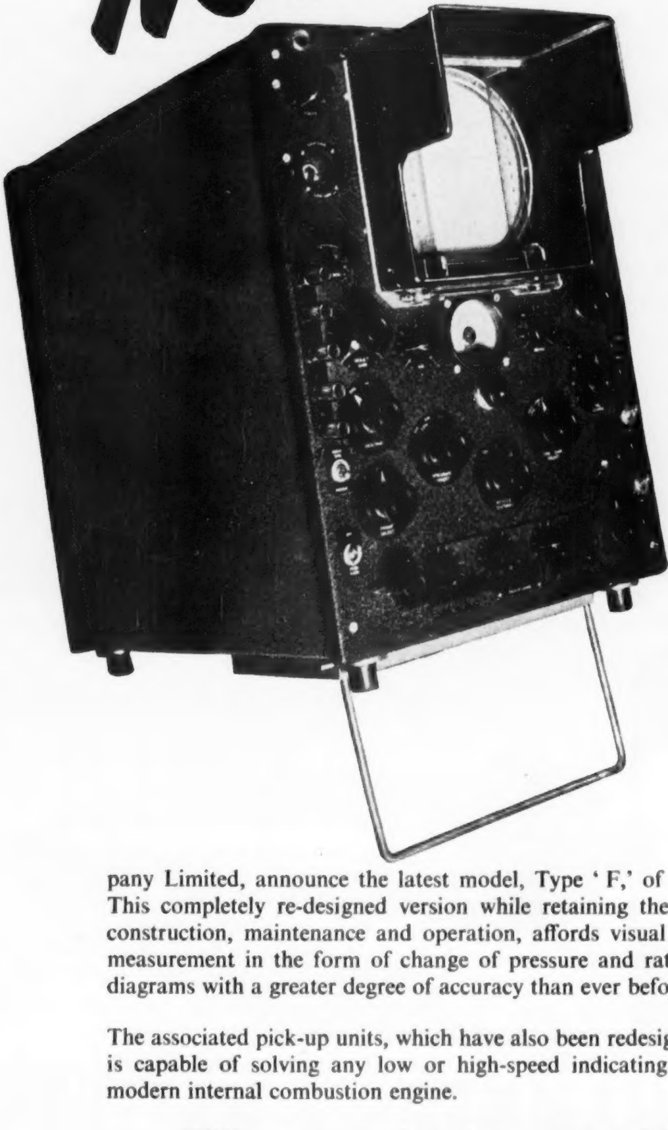

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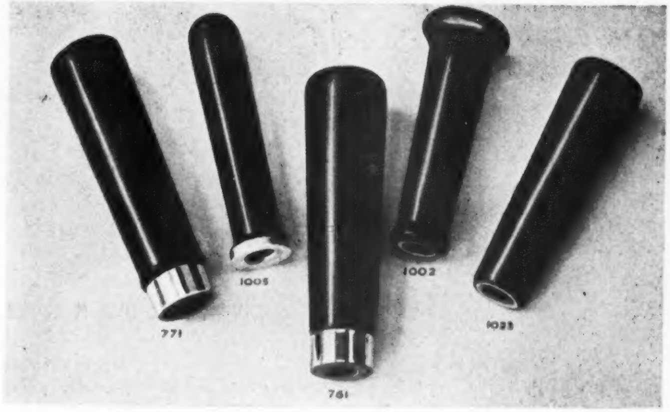

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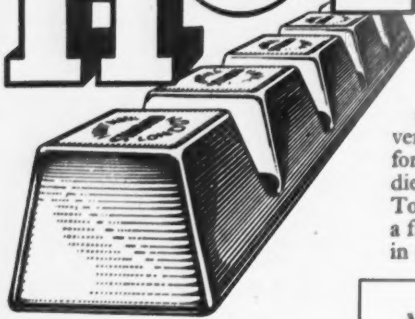

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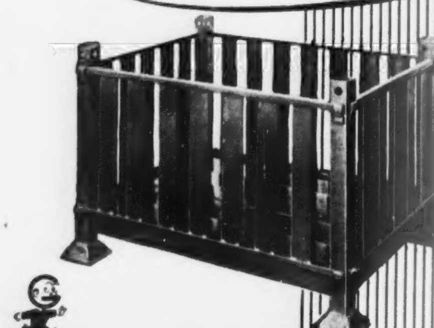

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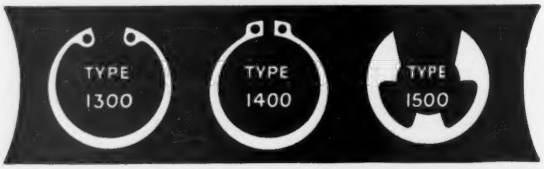

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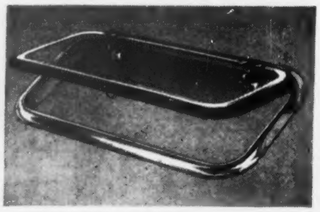

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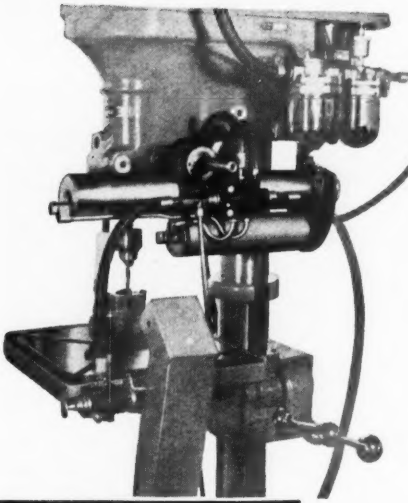

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